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CONTENTS.

PAGE.

METHOD OF RUNNING THE LINES FOR THE SHAFTING AND BORING OUT THE STERN TUBES AND BRACKETS OF THE U. S. S. *CINCINNATI*. By Geo. H. Kearny, Passed Assistant Engineer, U. S. Navy, ... 1

ALIGNMENT OF SHAFTING AND BORING OUT STERN TUBES, AS PRACTICED AT UNION IRON WORKS. By Leo Morgan, Esq., Member, ... 7

MARINE BOILER FURNACES. By D. B. Morison, Esq., ... 11

STEEL CASTINGS. By George D. Strickland, Passed Assistant Engineer, U. S. Navy, ... 39

ECONOMICAL SPEED AND COAL ENDURANCE OF WAR VESSELS, AS AFFECTED BY THE RELATION OF THE COAL EXPENDED FOR PROPULSION TO THAT EXPENDED FOR ALL OTHER PURPOSES. By Ira N. Hollis, Passed Assistant Engineer, U. S. Navy, ... 66

THE CONTRACT TRIAL OF THE U. S. S. *MONTEREY*. By Passed Assistant Engineer Howard Gage and Assistant Engineer Emil Theiss, U. S. Navy, ... 115

THE GREAT LAKES OF NORTH AMERICA. By George H. Ely, C. E., . . . 140

COMMERCE ON THE GREAT LAKES, ... 153

THE THORNYCROFT BOILERS OF THE DANISH CRUISER *GEISER*, ... 163

NOTES—

- High Smoke Pipes, ... 175
- On the Applications of Electricity in the Royal Dockyards and Navy, ... 178
- Electric Lighting, ... 183
- The Repairing of the *Umbria's* Shaft, ... 185
- Broken Crank Shaft of S. S. *Trave*, ... 188
- Classified Speeds of British Naval Vessels, ... 190
- Aluminum, ... 191
- Scholt and Gens' New Composite Gauge Glasses, ... 192
- Large Pillar Crane, ... 193
- Portland Cement for Steam Joints, ... 194
- Experimental Tank for Russia, ... 194
- Basic Bessemer Steel, ... 196
- The Peck-Wheeler System of High Temperature (Combined Live and Exhaust Steam) Feed-Water Heating, ... 196

SHIPS, ... 201

MERCHANT STEAMERS, ... 233

BOOK REVIEWS, ... 251

ANNUAL MEETING, ... 253

OBITUARY, ... 255

BY-LAWS, ... 256

OFFICERS AND MEMBERS AMERICAN SOCIETY OF NAVAL ENGINEERS, ... 259

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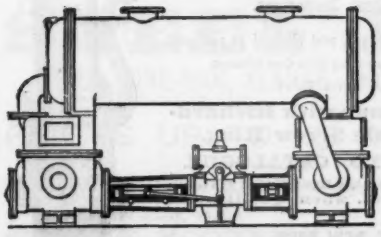

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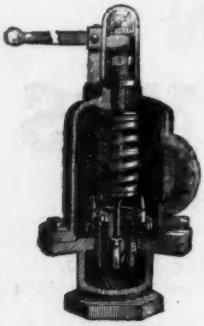

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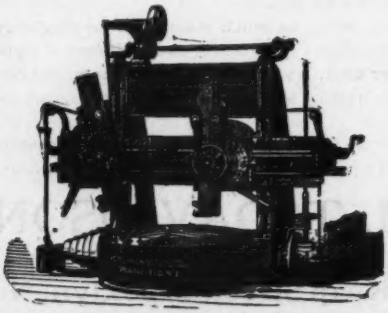

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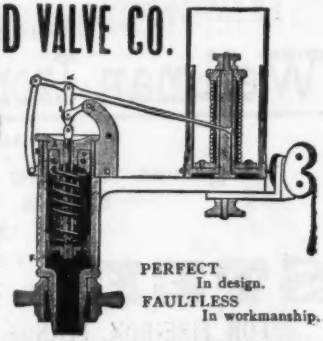

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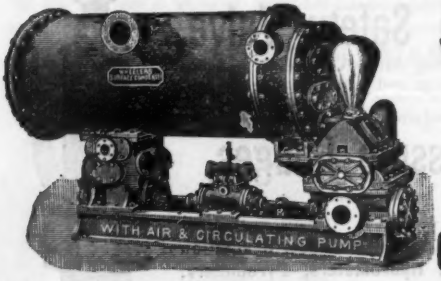

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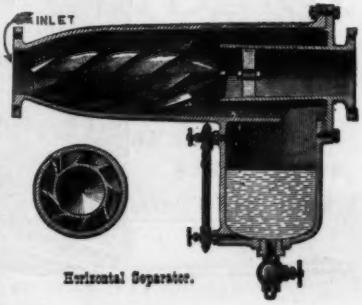

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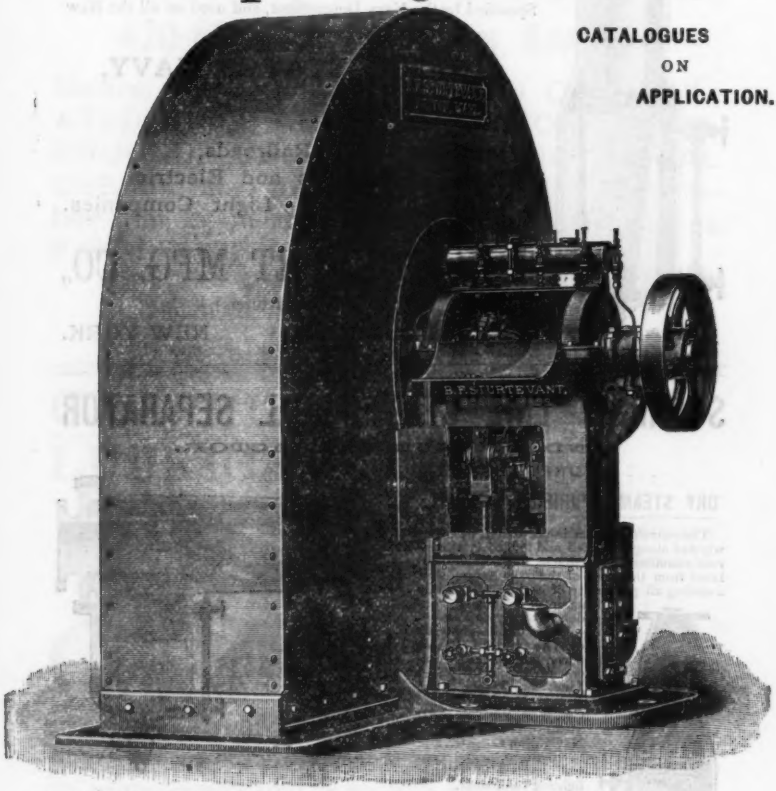

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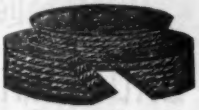

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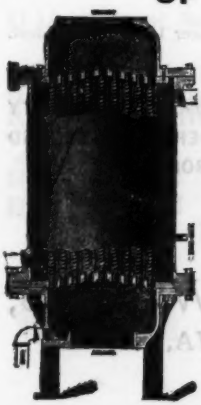

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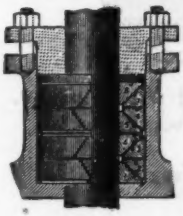

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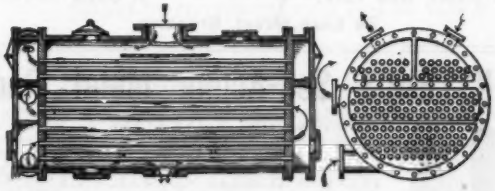

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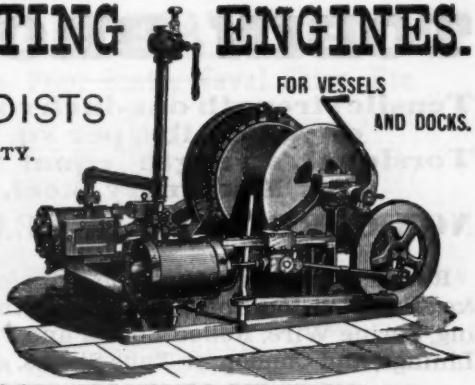

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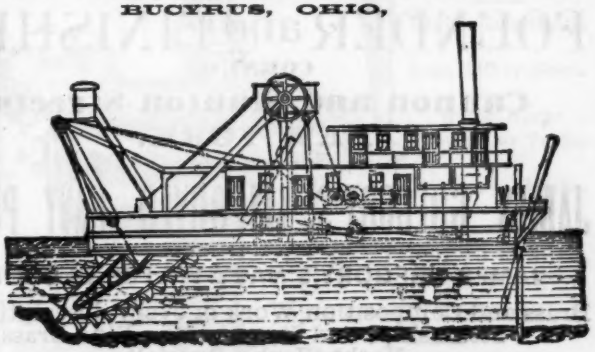

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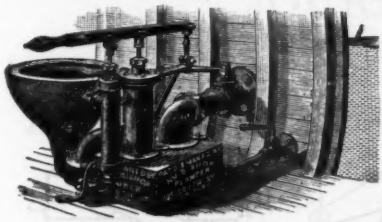

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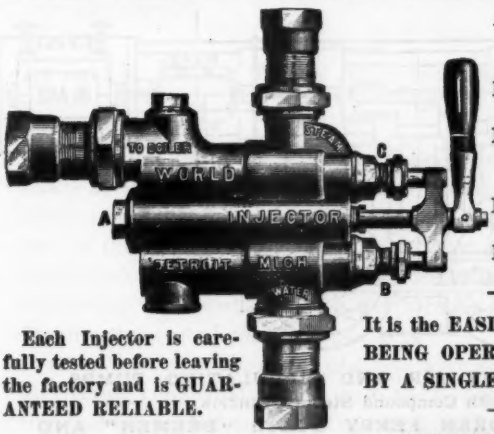

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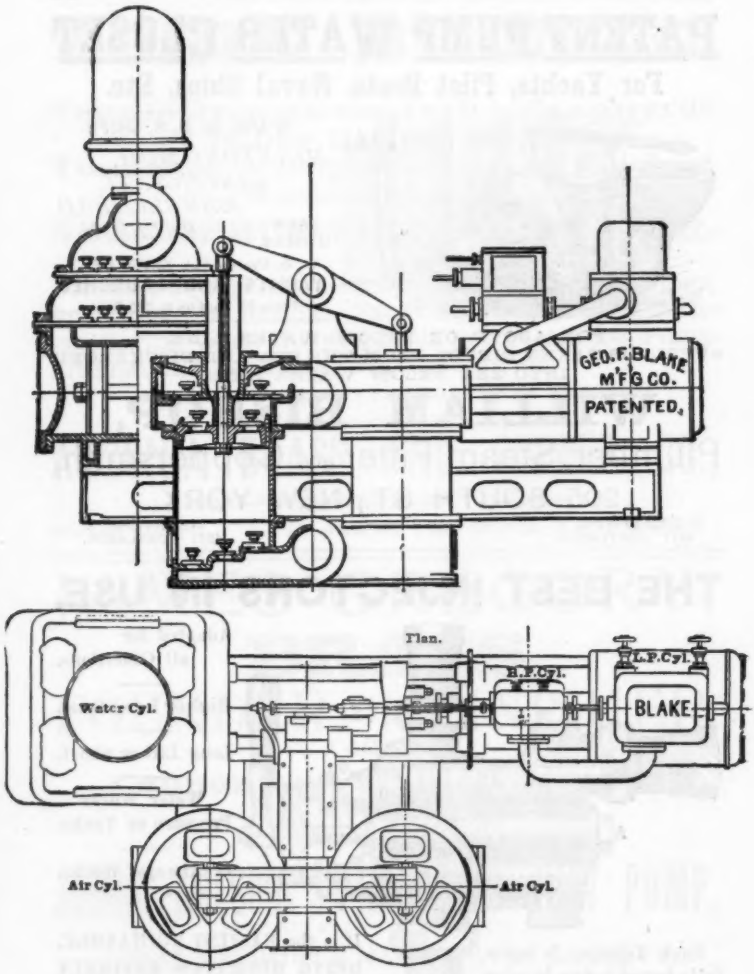

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JOURNAL

OF THE

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METHOD OF RUNNING THE LINES FOR THE SHAFTING AND BORING OUT THE STERN TUBES AND BRACKETS OF THE U. S. S. *CINCINNATI.*

BY GEO. H. KEARNY, PASSED ASSISTANT ENGINEER, U. S. NAVY.

The engine drawings gave the positions of the center of the shafts at the H.P. cylinders, and in the middle of the length of the stern brackets, above the base line and outboard of the center line of the ship. These dimensions were checked off on the mould-loft floor, and similar dimensions taken for points on the thwartship bulkhead, about $3\frac{1}{2}$ feet forward of the engines, and for the after end of the stern brackets.

After the stern tubes and brackets were secured in place and all riveting completed around them, the Constructor had the base line laid off on the outside of the hull, and from this base line and the mid-line of the keel were determined the shaft-line centers at the after side of the stern brackets and on the bulkhead

forward of the engines. This was done by means of straight-edges, levels, rules and plumb lines, the forward points being obtained by working up through openings in the hull in the forward part of the engine rooms for two of the outboard valves. A $\frac{3}{32}$-inch-diameter hole was drilled through the bulkhead at the forward end of each line, and a composition bushing fitted to it, this bushing having a $\frac{1}{64}$-inch-diameter hole in it. On the after side of the stern brackets, circles were struck from the centers found and marked on the steel casting. The distance between these end points is 115 feet 6 inches; the length of the stern tube is about 34 feet 10 inches, and from the after end of the stern tube to forward end of the stern bracket is about 27 feet 8 inches.

A steel wire was next stretched through these end points from the center point in the bulkhead to the wooden trestle work abaft the stern brackets, and kept in place for several days in order to determine what effect the sun had in warping the hull. Pointers were attached to the brackets and after ends of the stern tubes, and measurements taken from them on a wooden platform built up from the ground.

The early morning sun struck the hull on the port quarter, but by 11 A. M. it was off that bracket and stern tube, and after that time did not strike on the brackets till late in the afternoon, when it struck on the starboard bracket.

The measurements were taken in August, and it was found that on rainy or cloudy days but little movement was perceptible, but on bright days there was considerable lateral and vertical movement of the brackets, the port one showing the most.

Observations made at 5 A. M. and at 6.30 P. M. on a bright day gave the same positions for the stern tubes and brackets, and at 8 A. M. the change was hardly perceptible. Observations were made at 8 A. M., noon and 5 P. M.; these showed that the port bracket moved $\frac{1}{16}$ inch in height and $\frac{1}{32}$-inch laterally during the day. The movement of the end of the stern tube was about one-half that of the bracket.

The movements of the hull were plainly shown while boring out the stern brackets by the cuts taken just before and after the

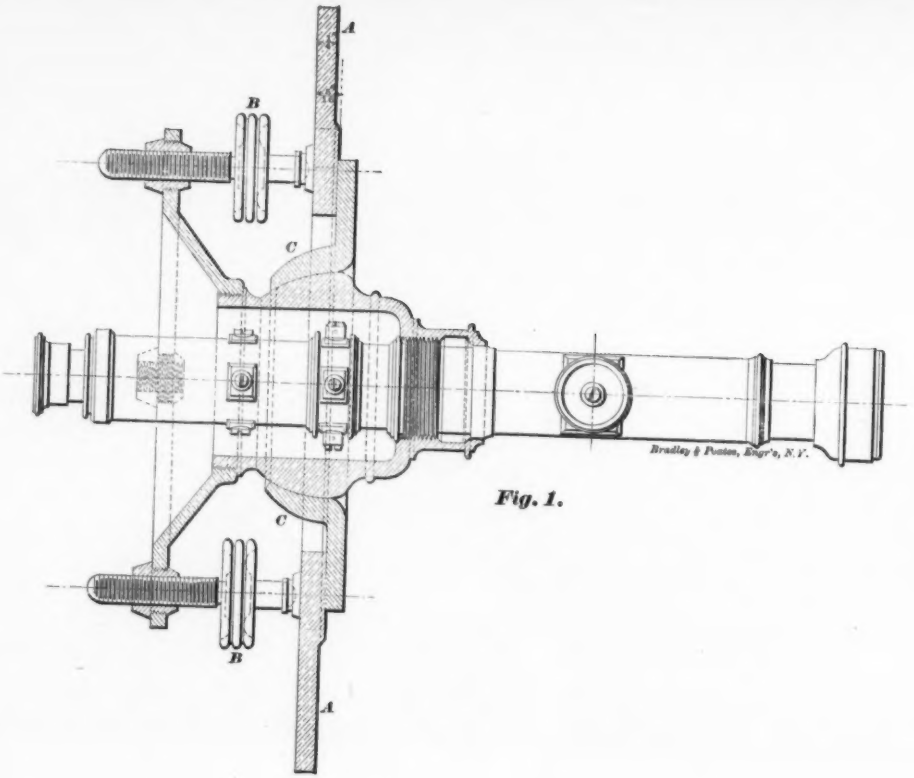


Fig. 1.

dinner hour, the boring bar being supported on a wooden frame work built up from the ground.

The finishing cuts were taken on cloudy days.

A better plan of securing the boring bar would be to support it from the hull. This was not done, as the plating was all on around the strut, and it would have necessitated drilling numerous holes in the plating, to be able to securely support a frame work.

For the purpose of locating the shaft-line centers, a transit telescope was bought and fitted up as shown in Fig. 1, the center of the ball joint, marked *C*, being made to coincide with the center of the telescope crosshairs, so that once adjusted and secured in place, the intersection of the crosshairs would remain in the center line of the shaft, no matter how the telescope was pointed. The telescope was secured, by means of four holes passing through the plate *A*, to a wooden ring wedged into and flush with the after end of the stern bracket; on this ring was described a circle, whose center was coincident with that of the circle previously described on the after face of the bracket, which was used to adjust the telescope in the center line of the shafting; this was done by slacking off two of the adjusting screws *BB*, and moving the plate *C* till a pointer gauge, fitting on the outside of the ball joint, showed the intersection of the crosshairs to be in the center of the circle drawn on the wooden disc. Then, by means of the screws *BB*, the telescope was adjusted so that the intersection of the crosshairs exactly covered the $\frac{1}{64}$-inch-hole in the bulkhead forward of the engines, a candle being held so that its light passed through the hole and was distinctly seen through the telescope.

In first running the lines, two pieces of tin were used, having a V-piece cut out of each, so that, by overlapping them, the opening could be gradually closed, till a very small opening was left, through which the ray of light could be seen. This process was slow and vexatious, as, in closing the slides when the opening was less than $\frac{1}{4}$ inch, the ray of light would sometimes be shut out, necessitating opening the slides again and repeating the process. Also, after reducing the opening to $\frac{1}{64}$-inch square

it was found that this small opening could be moved in any direction nearly $\frac{1}{16}$ inch and still show a glimmer of light through it. Blacking the edges of the slides did not correct this, so, finally, the following method was adopted:

A target of the shape and size shown in Fig. II was made; the right-angled center lines marked on it at the points *aaaa*; a piece of thin tracing paper cut to fit this circle, had drawn on it right-angled diameters, and was fitted to the points *aa*, and then clamped in place by the ring *bb*. The cover *e* had a very small hole at its center that served for centering the paper and also as a center for using trammels to describe circles on the ends of the stern tubes, &c., thus locating points in the center line of the shaft. Small holes *dd* served to secure the target by thumb tacks when it was in the right position.

Wherever a center point was to be established a piece of pine board was secured, and at the center given by the wire a one-inch hole was bored.

In using the target, the telescope was first secured and centered at the after end of the stern strut, then adjusted so that the crosshairs intersected on the $\frac{1}{64}$-inch hole in the bulkhead forward of the engines and clamped fast by means of the adjusting screws. The target (uncovered) was held in front of the one-inch hole in board center and a light held behind the board, bringing the cross lines distinctly into view; then the telescope was focussed on the target, and a minute's work sufficed to center the target, so that the telescope crosshairs would exactly cover the cross lines on the target. The target was then secured in place by thumb tacks, another sight taken to see that it had not been moved, and, if correct, the cover was slipped on and a pair of trammels used to describe a circle from this center on the previously-chalked surface of the steel castings of the stern tube or engine framing. Center points were thus established at each end of the stern tubes.

For the engine seatings, a box was made to fit in the pocket of the framing, where it was wedged fast, so that a one-inch vertical slot through the top of the box was at the approximate center given by the wire; then the target was used to give the exact

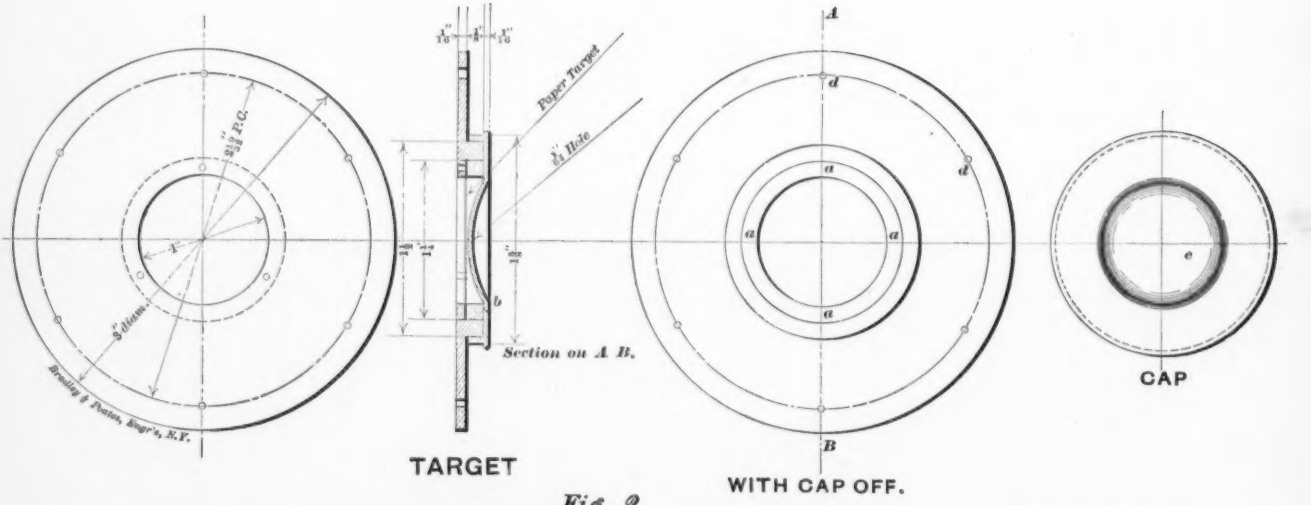


Fig. 2.

center. Centers were thus located for both ends of each of the four bed plates of each engine, and points established on the seatings.

The telescope would not focus for the forward end of the stern bracket, at a distance of only about 3 feet 6 inches from the object glass, so a line was stretched between the centers at the after end of the bracket and the one in the after end of the tube, and, allowance being made for the sag of the line, a circle was described on the forward face of the bracket to assist in locating the boring bar.

The cross lines on the target were about $\frac{1}{90}$ inch wide, and if the target was $\frac{1}{100}$ inch out of center it could be detected. This was proved by twice striking in three different centers, and in each case having the first and second found centers coincide exactly, as shown by describing circles on the wooden center pieces.

The work of establishing these points was done either before 7 A. M. or after 5 P. M., when there was no appreciable movement of the hull.

This method is very quick and easy, and is perfectly accurate, provided the telescope is in perfect adjustment; but care must be taken that the line of collimation of the telescope is perfect, otherwise in focussing at the different distances the line of sight will be thrown out of the true line and wrong centers established. Of course, any error in the telescope will affect any other method as much as this one, where the telescope is focussed for the different distances. Horizontal and vertical lines were also established at each end of the engine seatings by means of straight-edges, levels and plumb lines, for use in placing the bed plates and erecting the engines.

Before the boring bar was placed in position, a line was stretched through the tubes, and exactly centered at the ends and in the middle; then the tube was carefully callipered, to make sure there was sufficient metal to bore out for the seatings. In the starboard tube it was found necessary to move the shaft center line down and inboard $\frac{1}{16}$ inch. After this was done all the intermediate center points were again established.

The boring bar used was hollow and only 24 feet 9 inches in

length, and therefore had to be shifted in order to bore out the seatings for the three bushings in the stern tube. In boring out the stern brackets, one end of the bar was set by the circle described on the after face of the bracket and the telescope placed in the after end of the bar, the plate *C* having been made of such diameter that it exactly fitted into a recess in the bar, thus centering the telescope crosshairs. At the other end of the bar, in a similar recess, was fitted a ring, across which were stretched two fine horsehairs intersecting at the center of the bar, and then the forward end of the bar was moved till the telescope crosshairs and those in the forward end of the bar were in line with the $\frac{1}{64}$-inch hole in the bulkhead forward of the engine room. The boring bar having been thus located, its end bearings were secured and the center bearing adjusted so as to compensate for the sag of the bar that had been previously determined. In boring out the stern tubes, the bar was first placed in the after end, and the after end of the bar set by means of the circle previously established on the after face of the tube; the telescope and crosshairs again placed in the bar and the forward end of the bar adjusted by means of six bolts working in the shell of the stern tube, just forward of the seating for the middle bushing, and working against a composition sleeve in which the boring bar revolved. The sag of the bar was taken up at its middle by another sleeve which was also adjusted by bolts fitted through the shell of the stern tube.

After boring out the seatings for the after and middle bushings in the stern tube, the bar was slipped through the tube, till its forward end projected beyond the forward end of the tube where it was centered by the circle struck on the forward face of the tube, the after end being centered by a bushing fitting the seating already bored out for the middle bushing, and the sag of the bar compensated for by a sleeve at its middle of length.

The boring bar was also used to face off the ends of the stern tubes and brackets, after which the bushings and fair water ine pieces were put in place and centers marked for the studs which were screwed into the steel castings to secure the bushings, etc.

The middle bushing of the stern tube was connected to the after-bushing by two composition sleeves and all screwed together. These were all drawn and shoved into the stern tube and, when in place, four set bolts, tapped into the stern tube casting were let into the middle bushing to prevent its turning with the shaft. The forward bushings in the stern tube and those in the brackets are in halves, to enable their being more easily removed.

ALIGNMENT OF SHAFTING AND BORING OUT STERN TUBES, AS PRACTICED AT UNION IRON WORKS.

By Leo Morgan, Esq., Member.

The following article is a description of the method of lining up and boring out the stern tubes of Cruiser No. 6, U. S. S. *Olympia*, now building at the Union Iron Works, San Francisco, Cal.:

On forward bulkhead in engine compartment, lay off from the vertical center line of vessel the distance required to centers of shafts; from engine seating lay off on same bulkhead the distance from seatings to centers of engine shafts; the intersection of these measurements will locate the centers of forward ends of shafts. Through these centers drill a small hole $\frac{1}{4}$ inch in diameter and we will have our first "fixed point."

Proceed to after bulkhead in engine compartment, and lay off in a similar manner the approximate centers of shafts and cut a hole somewhat smaller than the diameter of shaft, to be enlarged to size required by drawing, after the center line of shafting has been accurately determined. Insert in after end of propeller-shaft bracket a piece of board 1 inch by 4 inches, having a $\frac{3}{4}$-inch hole approximately central, and faced on after side with a piece of 4-inch by 4-inch light Russia iron; find center of bracket,

from center punch marks previously located, and drill a hole $\frac{1}{16}$ inch in diameter through the Russia iron. This will give us our second "fixed point."

With these two points all intermediate bearings must come accurately in line. Having placed a light on forward side of the $\frac{1}{4}$-inch hole in forward bulkhead, we can now proceed to "sight up" the different points required. Beginning at the after bulkhead in engine compartment, place a piece of 1-inch board of suitable length, 4 inches wide, having an oblong hole 1 inch by 3 inches in the center, across hole in bulkhead, and adjust till the light, viewed from center in propeller shaft bracket, comes approximately in center of oblong opening, then bolt board fast to bulkhead.

Place a piece of 1-inch by 6-inch light Russia iron horizontally across the oblong opening in board and move it vertically upwards until the light, viewed as before, is completely shut out; lower the piece until the light is again wholly exposed, then secure the strip in place; parallel to this and $\frac{1}{16}$ inch above it, secure a similar piece.

Over these two pieces place another 1-inch by 6-inch strip and move it across the $\frac{1}{16}$-inch opening until the light is shut out as before, move backwards until the light is again uncovered and secure strip in place; parallel to this and $\frac{1}{16}$ inch distant from it, secure another 1-inch by 6-inch strip. We now have an opening $\frac{1}{16}$-inch square, through which the light from the forward "fixed point" should show clearly.

Place a piece of 4-inch by 4-inch light Russia iron with a $\frac{1}{16}$-inch hole in its center over the four 1-inch by 6-inch strips, bringing the hole in the 4-inch piece central with the $\frac{1}{16}$-inch square hole in strips; secure the 4-inch by 4-inch piece firmly in place, and remove the four 1-inch by 6-inch pieces.

Next proceed to forward bearing *C* in stern tube, place a piece of 1-inch by 4-inch board with a 1-inch by 3-inch hole in its center, in bearing, and find exact center in the manner first described.

In this manner treat each bearing in succession, and when finished, if due care has been taken in the work, the light from

forward bulkhead will show clearly through all the centers erected between our two "fixed points." If the light does not show through the several centers they are removed, one after another, beginning with the center last placed in position, until the one in error is reached, when the process of recentering is gone through with. In a similar manner the exact center in ¼-inch hole in forward bulkhead can be determined. As each bearing is centered, its center is located with reference to four center-punch marks on face of bearing, and a circle, equal in diameter to bore required, described and center punched to ensure location.

If we find that any of the bearings will fail to "clean up" in boring, the center in propeller-shaft bracket, or that in forward bulkhead in engine compartment, can be changed to ensure the cleaning up of all the bearings.

During this process, the alignment of the engine seatings can be determined by erecting, at three or more points, pieces of boards having a 1-inch by 3-inch hole in top ends, bringing these holes in line with light, viewed from center in after bulkhead in engine compartment, and fixing light Russia iron centers as before described.

The "sighting up" can be done best at dusk or after night has set in.

After the centers have all been determined, the 1-inch by 4-inch pieces are removed from the after bearings in stern tube and the bearings *BB* for the boring bar placed in position; the center in after face of bracket re-inserted, a line passed through it and center in forward end of stern tube for the purpose of roughly centering the bearings *BB*. The boring bar is then slipped into place; the boring gear placed on its platform and the end of bar inserted. The bar is now accurately centered with reference to the center punch marks on face of bearings *CCCC*, and after face of propeller shaft bracket, and its bearings securely fastened. Owing to the shortness of the bar, only four bearings could be finished at one setting.

To bore the forward bearings, the boring gear had to be shifted to forward side of stuffing-box bulkhead, the bar placed in position, centered with reference to center punch marks on the

face of bearings and secured as before. For boring out the forward bearings a hole was put through stern tube to allow worm spindle to project outside. At each setting the bearings are all roughed out to within $\frac{1}{16}$ inch of required size before finishing.

After all the bearings have been finished to size, the sights are again placed in position and the accuracy of the work tested. If the alignment is satisfactory, the after face of propeller-shaft bracket is faced off; the holes for securing studs and bolts drilled and tapped, and the stern-tube bearings placed in position and secured. The boring should be commenced on the side from which the stern-tube bearings are entered.

The staging and platform *P*, previously erected, for carrying the outer end of bar and gear is shown in Figs. 1 and 2. The straps for supporting the forward end are secured to stern tube, and made of 1-inch by 3-inch flat iron, with $1\frac{3}{8}$-inch round ends welded on.

A $1\frac{3}{8}$-inch round iron strap secures it to after end of bracket, and the after end is supported by two straps suspended from hull. Lateral stays are also fitted to convenient points.

Figure 3 is a detail of boring bar.

Figure 4 shows the bearing *BB* for supporting boring bar.

The apparent complexity of this bearing is accounted for by the fact that this bar and fittings were designed for boring out the stern tubes of the *San Francisco*, the four extensions or feet being added to suit enlarged diameter of stern tube.

If we wish to test the accuracy of the center in bracket, we may proceed as follows: Level out from the short line laid off on hull (this line is parallel to base line of ship and at a known distance from it) to the after face of bracket; the distance from center in bracket to this line, plus the distance of this line above base line of ship, will give us the vertical distance required. Across the after side of stern post and square with center line of keel place a straight-edge; square out from this point to center of stern bracket will be the horizontal distance required.

The motive power for operating the bar can be furnished by a small portable engine or any other suitable device.

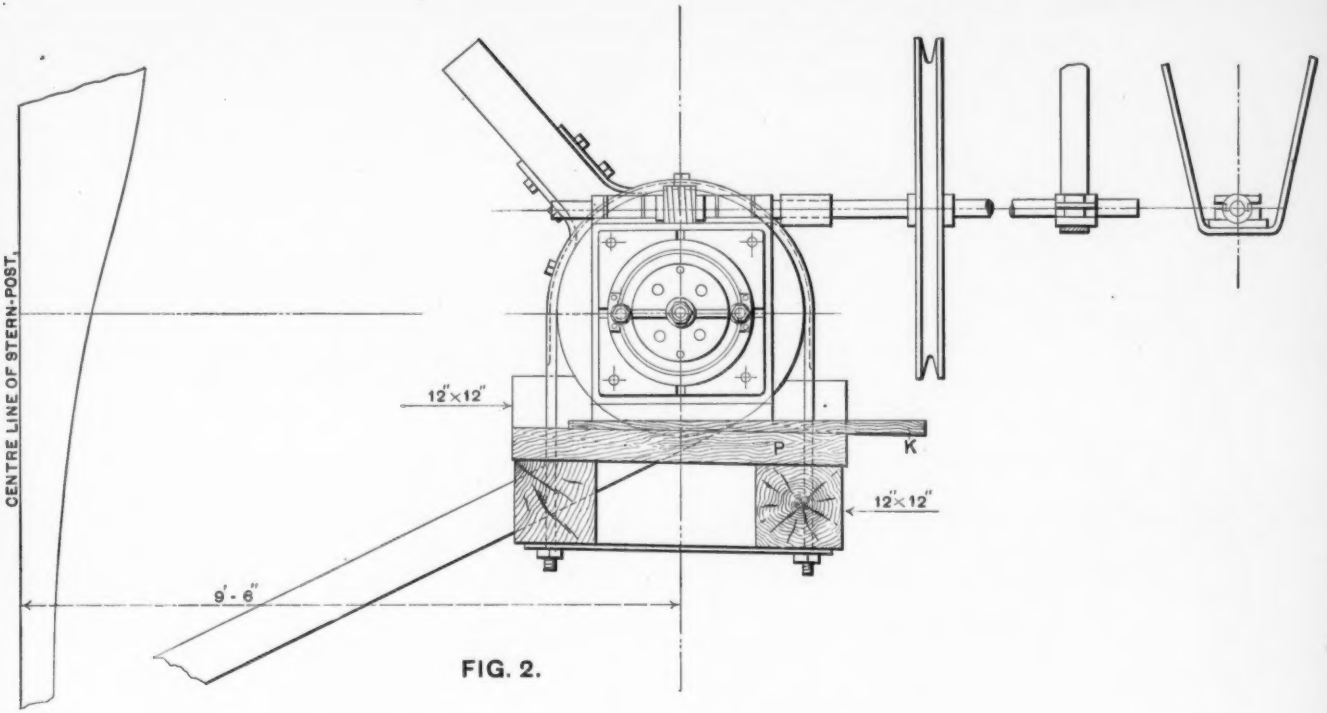


FIG. 2.

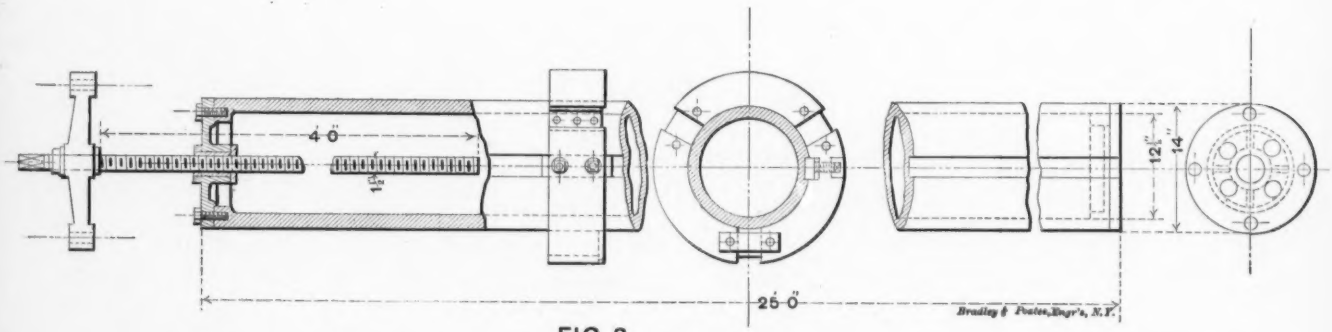
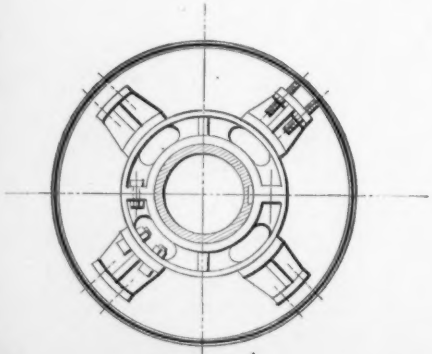


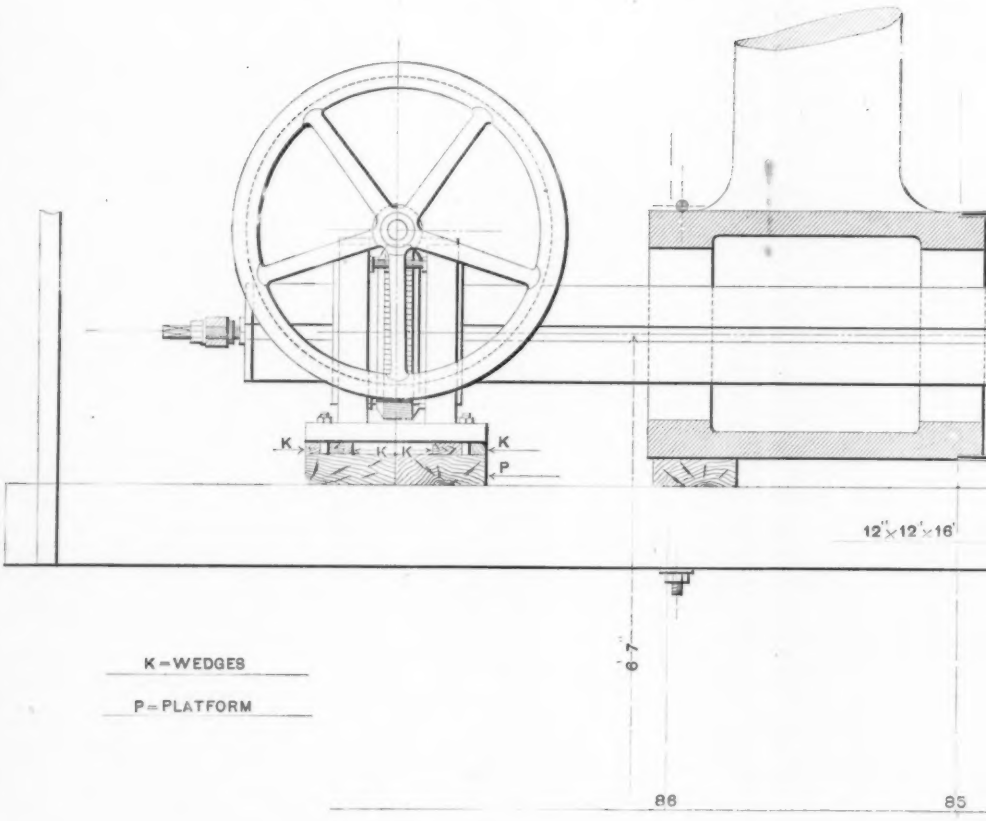
FIG. 3



SECTION A B.

FIG. 4

XU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12"×12"×16'
K=WEDGES
P=PLATFORM
6-7
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SCALE: $\frac{1}{8}$"=1'

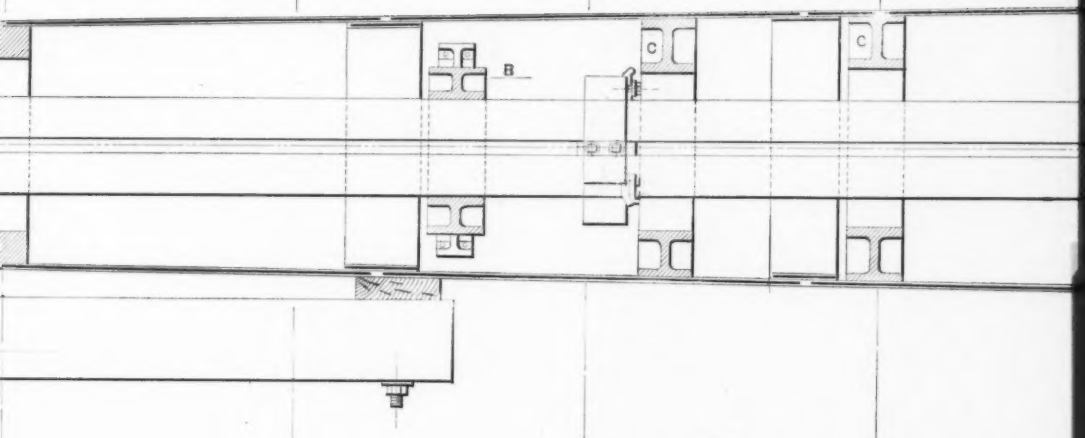


FIG. I

5 84 83 BASE LINE 82

SCALE: $\frac{1}{8}$"=1'

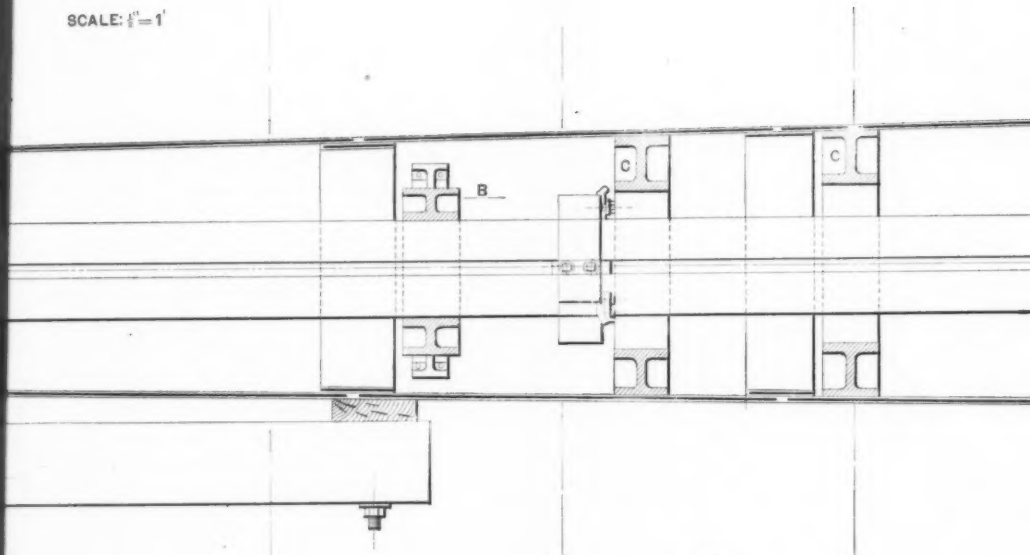
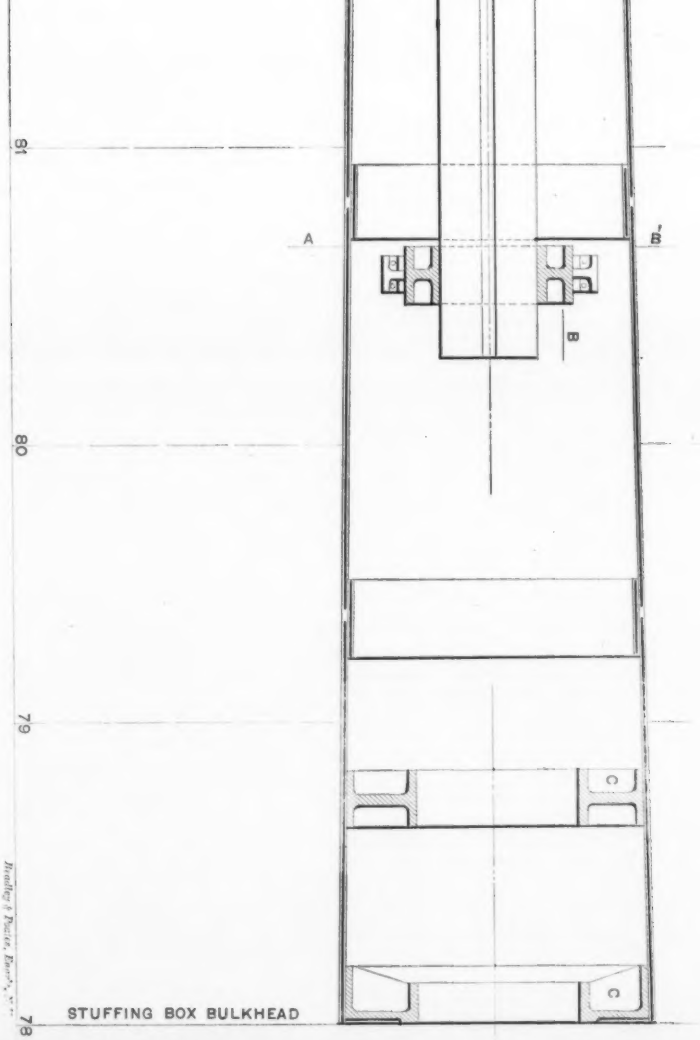


FIG. I

84 83 BASE LINE 82


91
80
79
78
A
B
B
B
C
C
STUFFING BOX BULKHEAD

MARINE BOILER FURNACES.

By D. B. Morison, Esq.

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A boiler furnace is one of the most important details in marine engineering, as it not only represents the most efficient heating surface in the boiler, but, in the event of accidents, its replacement or repair is highly expensive in itself, and involves the temporary disablement of the steamer.

From the nature of the manufacture and the great expense of experimental tests, engineers generally have had less opportunity of becoming thoroughly conversant with modern furnaces than with any other detail of marine engineering, and the object of this paper is to lay before the Institution the results of some experiments which have been made, and also by advancing certain theories resulting from actual experience to arrive by discussion at the probable cause of the many commercially disastrous accidents of recent years.

The chief points upon which this paper will treat are: Design mode of manufacture, practical requirements, strength and material.

As soon as steam of a higher pressure than that of the atmosphere began to be used it was found desirable to replace the ordinary square section furnace with its flat sides and top by a design theoretically stronger and less liable to collapse under pressure; consequently the cylindrical form was chosen and, with modifications, still remains in use. When steam pressure of 100 pounds per square inch and over became general, the thickness required for a long plain furnace was excessive, and engineers considered how it was possible to obtain the necessary strength in a more scientific manner than by increasing the thickness of the plate.

Experiments made by Fairbairn had shown that the strength of a flue under collapsing pressure varied with the length, and that short flues were stronger than long ones, so the first step taken to increase the strength of plain flues was by adding strengthening rings circumferentially, thus practically making one long flue into several short ones. At first angle or T rings were riveted on, but as the rivets were directly exposed to the fire, a better arrangement, known as Adamson's rings, was ultimately adopted. This design consists of short plain cylindrical tubes, flanged outward at the ends so that two rings or more may be conveniently riveted together, the rivets being completely covered by water, and so protected from the fire. The method of manufacture of flues constructed with Adamson's rings is so well known that further explanation is unnecessary, and although marine engineers prefer a furnace with no circumferential joints, these furnaces are still fitted in high-pressure boilers, and various experimental results, which have been kindly placed at the disposal of the writer by Messrs. Hall, Russell & Co., of Aberdeen, and Messrs. J. Howden & Co., of Glasgow, will be referred to subsequently.

A great step forward was the introduction by Mr. Samson Fox of his corrugated furnace flue, the strength of which is so much in excess of that of a plain flue that it came rapidly into use and was for a considerable time almost universally adopted for high-pressure boilers; indeed, it is certain that the rapid adoption of the triple-expansion engine was due in a great measure to this invention. Corrugated furnaces were first introduced in 1879, the method of manufacture being the crude one of hammering a plain flue on a corrugated anvil block. In 1882 Mr. Samson Fox invented his rolling mill, which is still used by the Leeds Forge Company for the production of both Fox's corrugated and Morison's suspension furnaces, the latter being an improved design lately introduced. These furnaces are made from Siemens-Martin steel ingots rolled into plates of the required dimensions under ordinary plain rolls. Three sides of a plate are sheared, and on the fourth side the development of the saddle is marked and punched out. Test pieces are taken from the

scrap thus produced and tested for tensile strength, elongation and bending. The plate is then formed into a tube and lap welded by water gas, the welding by this process being so efficient that in a series of tests lately made the tensile strength across the weld was found to be but slightly less than the original plate. This welded tube is next heated in a special furnace and is then placed in Fox's patent mill, the rolls of which are corrugated, and in order to get the tube between the rolls the top roll is worked out and in longitudinally by hydraulic power. Complete corrugations are formed by one revolution, but a few turns are given for finishing, and the furnace is withdrawn as a perfectly cylindrical corrugated tube. Flanging is the next process, and, although this has hitherto been done by hand, arrangements are being made for flanging by hydraulic pressure. This undoubtedly will be a great improvement, because of the fact that the greater the amount of work put on the back ends of furnaces of any type, the greater is the liability to crack, especially if the steel is of very high tensile strength. The furnace is now complete, the final process being annealing.

A modification of the "Fox" is that known as the "Farnley" flue, the corrugations of which are similar to those of the Fox, but instead of being at right angles to the axis they form a series of spirals, the advantage claimed being additional longitudinal rigidity.

Another step in the direction of increased practical efficiency is a design by Mr. Holmes, of Hull, the feature in this furnace being that the outward corrugations are pitched about 16 inches from center to center, the portions between the corrugations being of plain cylindrical section. In the Holmes furnace, the corrugations are 2 inches in depth, and are formed one at a time by pressing the metal between suitable dies, the furnace being revolved between each application of pressure.

The furnace which has been the greatest rival to the Fox is that known as Purves' patent, and is manufactured by Sir John Brown & Co., of Sheffield. This furnace was patented in 1880 by Mr. Purves, late of Lloyd's Registry, and consists of a series of thickened ribs 9 inches between centers, the part between the

strengthening ribs being of plain cylindrical form. A novelty in this furnace is the method of manufacture, as it is the first furnace of unequal section, and the first furnace not made from an originally plain plate of equal thickness throughout. The "Purves" flue is made from Siemens-Martin ingot. Rectangular section slabs sufficient for two flues are formed from these ingots under a 15-ton hammer, the slabs being about $7\frac{1}{4}$ inches thick, and their length being equal approximately to the length of the flues required. Special roughing rolls convert the slab into a ribbed plate $1\frac{1}{4}$ inches thick, which is then cut in two by very powerful shears, and, after re-heating each half is passed through finishing rolls until the final required thickness is obtained. At one side edge there is a piece of plain plate 15 inches wide to the center of the first rib intended for the front end of the flue, and at the other side the plain part is 23 inches wide and intended for the back end of the flue, and the thickness in both these side edges is increased by one-eighth to allow for the thinning which takes place during flanging, &c. After being sheared at the edges and the end slotted to the shape required, the plate is bent into a circular form by a special hydraulic press. The edges are then welded together by the insertion of glut pieces, the plain parts being welded first and the ribs afterwards. Annealing is the next process, and after being withdrawn from the furnace the flue is converted into a perfectly circular tube by a very ingenious hydaulic press. It is then flanged, &c., in the ordinary way, the final process being annealing.

The latest design introduced is that known as Morison's suspension furnace, which is a modification of both the Fox and Purves types, and a combination of the good features of each. It is manufactured in exactly the same way as the Fox, the same processes being employed throughout.

An ingenious arrangement (Fig. 15) for facilitating the removal and replacement of furnaces (which are flanged to meet the combustion-chamber tube plate) has been devised by Mr. Ashlin, and consists in cutting away the lower half of the flanged end, so that it terminates in a plane inclined at about 120 degrees to the

SECTION OF MORISON'S SUSPENSION FURNACE
IN TESTING APPARATUS.

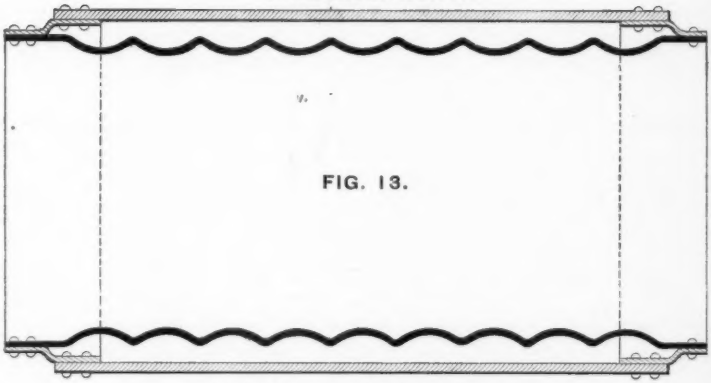


FIG. 13.

1891 BOARD OF TRADE EXPERIMENTS.

MORISON'S SUSPENSION FURNACE.
ARRANGED FOR REMOVAL ON ASHLIN'S PLAN.

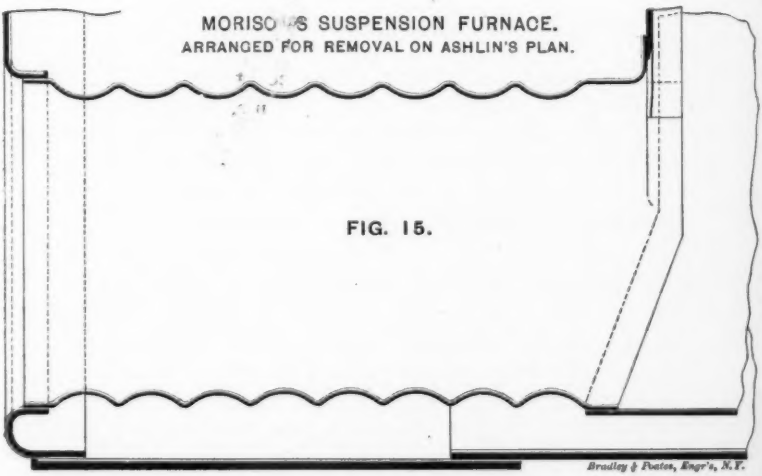


FIG. 15.

Bradley & Pates, Engr's, N.Y.

horizontal, and consequently the furnace, when being removed from the boiler, can be raised and tilted at its front end, thus enabling the flanged portion to be withdrawn.

REQUIREMENTS OF A FURNACE.

The apparent practical requirements of a furnace suitable for high pressures in modern marine boilers are—

(*a*) The furnace should be such as to give it the greatest evaporative efficiency and the least mean temperature.

(*b*) The material should be so disposed as to give the greatest resistance to collapse without undue rigidity in a longitudinal direction.

(*c*) The strength of the flue should be uniform throughout its length, so as to prevent partial collapse or local sagging.

(*d*) The material should be of equal thickness, so that expansion may be uniform and not unequal and local.

(*e*) The formation should be such that the strains resulting from extreme variations of temperature should be distributed throughout the length of the furnace, and not concentrated at any point so as to induce cracking.

(*f*) There should be no narrow cavities or recesses for undue and unequal accumulation of deposit, resulting in the burning of the material, and also in decreased evaporative efficiency.

(*g*) The strengthening projections should be outwards, or towards the water space, and so protected from the fire.

(*h*) The furnace should be of such formation as can be easily scaled, cleaned, repaired or replaced.

(*i*) The material of which the furnace is made should be such as has been found by experience to be reliable in manufacture, and the most suitable for withstanding the extreme variations of temperature, rather than such as has the greatest tensile strength. In other words, the material should be of the best quality for standing the comparativley rough treatment of the boiler shop, and the alternate and repeated heating and cooling of the furnace when in use, and the necessary strength should be obtained by the efficient disposition of material, thus giving suitability of

material the first consideration rather than the strength of material.

Before proceeding to consider how the furnaces under notice fulfill the requirements previously enumerated, it should be noted that the minimum steam pressure in a modern marine boiler is 160 pounds per square inch, and the tendency is to increase that pressure up to 180, and even 200 pounds, and also that a furnace of large diameter with a short grate is more efficient than a small diameter with a long grate; for two reasons —one being that the combustion is more perfect in the larger furnace, and the other—the practical reason—that an average fireman is able to fire a short grate better than a long one, especially in these days of limitation of work and disinclination to exert a maximum amount of energy. These considerations alone seriously detract from the value of the ordinary plain furnace, as, all other conditions being similar, it is the weakest form of cylindrical furnace at present in use; and, as the greatest thickness allowed by the Board of Trade is $\frac{5}{8}$ of an inch, the largest diameter obtainable for 180 pounds, with a furnace 6 feet 6 inches long, is 28.647 inches.

Lloyds, however, allow a greater thickness, and many engineers use plain furnaces up to $\frac{7}{8}$ of an inch; but the all-important commercial question is—Do these thick furnaces of small diamter result in the maximum obtainable efficiency in a modern steamship? And as the keen competition in shipping demands that for success attention should be given to every detail, it is necessary that a furnace should be not only the most economical for steam producing, but should be of such a formation as will last the greatest length of time, and have the least prejudicial effect on the wear and tear of the boiler. The material of a furnace is subjected to greater variations of temperature than in any other part of the boiler, being greatest in amount above the grate bar and least below. Intensity of temperature above the grate bar depends on the condition of the fire, the thickness of the plate, the amount of deposit on the plate, and the steam pressure. The variations of temperature are incessant, and depend on condition of fires, opening and shutting of doors, and

cleaning of fires. These variations of temperature result in expansion and contraction, producing definite mechanical movement, and if the design of furnace is such that it cannot readily adapt itself to those movements (and no plain furnace can), either the material becomes distressed, or such strains are produced on the boiler that leakage results. It is for this reason that excessive corrosion is often found at the grate bar level of plain furnaces.

The Adamson design is a considerable advance upon the plain furnace, from the fact that it is less rigid in a longitudinal direction and is stronger to resist collapse. It has, however, the objectionable feature of a series of circumferential riveted joints. This defect is removed in the Holmes furnace, which, although not introduced immediately after the Adamson, is the next step in advance of it.

The Fox furnace at the time of its introduction was the strongest form to resist collapse ever made, whilst the nature of its formation rendered it specially suitable for accommodating itself to the strains resulting from variation of temperature, and the fact that it is of uniform thickness and of uniform design caused the strains to be distributed throughout the length of the furnace. This uniform distribution of strains is the most important feature, as, if these strains are localized, accidents will ultimately result. Theoretically, the Fox furnace approaches perfection; practically, however, it has defects, the chief of which is the undue accumulation of deposit in the narrow cavity formed by the inward corrugations, and as these corrugations are nearest to the fire the material becomes unduly heated and frequently cracks. The scale in these narrow cavities is also difficult to remove, and often but imperfectly removed, thus intensifying the danger of accident. The general recognition by engineers of these practical defects of the Fox furnace was undoubtedly the chief cause of the rapid adoption of the Purves design, in which not only are the supporting ribs in the water space, (and so removed from the action of fire), but there is no cavity for the undue accumulation of scale, and no greater expense incurred in scaling than with a plain furnace.

These advantages are obtained, however, at the sacrifice of others, as, all conditions being similar, the Purves flue is not so strong as the Fox to resist collapse, whilst its unequal section and longitudinal rigidity localize the strains resulting from variation of temperature, an evidence of which is the occasional development of cracks either by circumferential ripping at the base of the rib or in the vicinity of the flanges at the back end, and, although this is an accident which is fortunately not common, it is sufficient to show the advantage of uniform distribution of strains. The section of the Purves flue being unequal, and the strength to resist collapse not being uniform, the first sign of collapse is the local sagging or pocketing of the flat parts between the ribs. The furnace, however, has been largely used, and it was the writer's practical experience with it which suggested the design of the suspension furnace, the object of which was to combine those features in the Fox and Purves designs which experience had shown to be necessary for practical success, and to reject those features in each which practice had proved to be bad. The features retained from the Fox design are, the disposition of the material in a form which gives the greatest resistance to collapse, uniform thickness and uniform strength throughout, distribution of strains uniformly throughout the length of the furnace. The features retained from the Purves flue are, the strengthening formations are in the water space and are protected from the fire, there are no narrow cavities for the accumulation of scale, and there are equal facilities for scaling and cleaning.

A feature often claimed for the Fox furnace is additional evaporative efficiency due to additional area of heating surface, but, when it is remembered that the current of heated gases impinges on the inward corrugations whilst the recesses formed by the outward corrugations are filled with gas comparatively stagnant, and also when it is remembered that the inward corrugations by receiving the greatest heat produce the greatest thickness of deposit, the efficiency of this additional surface is, in the writer's opinion, often overestimated. In the Purves design the heated gas sweeps over nearly the whole surface, the

only recesses being those formed by the ribs, and the same is practically the case in the Morison furnace. A frequently suggested arrangement is the simple reversal of the latter design, thus making the long suspension curve into an arch, but a little consideration will show that not only would the material forming the inward projections become overheated by the current of hot gas sweeping over the points, but the recesses formed by the arched surface would be filled with eddies of somewhat stagnant gas of a much lower temperature, and there can be little doubt that the evaporative efficiency of such a furnace would be less than a plain furnace of equal thickness.

A design which has been patented and re-patented is a furnace of barrel shape, either of equal thickness throughout or of varying thickness. The most successful furnace of this type at present being manufactured is that of Messrs. W. Arnold & Co., of Barnsley, and consists of a series of flanged barrel-shaped sections. This form is not suitable for marine work; and another design which has been proposed, consisting of a single length of barrel-shaped tube, would be still less so, and would be a very weak form of furnace.

There has been a host of other designs evidently patented with a view of evading recognized successful manufactures and without any idea whatever of actual improvement, but naturally none of these have been adopted.

STRENGTH TO RESIST COLLAPSE.

The first experiments upon the collapsing strength of plain cylindrical tubes were those by Fairbairn, but as the greater number of these were made upon cylinders of small diameter it is very doubtful whether the results obtained apply to flues of the dimensions in modern practice. Fairbairn's experiments have, however, been accepted by the majority of engineers as reliable, and the formula deduced therefrom has formed the basis of all rules for the strength of plain cylindrical flues. Fairbairn's formula was:

$$P = \frac{806{,}300 \times K^2}{L \times D}$$

where P = the collapsing pressure in lbs. per square inch.
K = thickness of the plate in inches.
L = length of the flue in feet.
D = diameter of the flue in inches.

This implies that the strength of a flue to resist collapse varies inversely as its length, and directly as the square of the thickness; but the latter may be questioned when the flue is of the proportions necessary for marine boilers. Both the Board of Trade and Lloyd's have adopted this formula, with an alteration of the constants.

Board of Trade (steel furnaces)—

$$\text{Working pressure per square inch} = \frac{99{,}000 \times K^2}{(\text{length in feet} + 1) \times D}.$$

Lloyd's Register—

$$\text{Working pressure in pounds per square inch} = \frac{89{,}600 \times K^2}{L \times D},$$

although there is a limitation in both cases that the pressure must not exceed $\frac{8{,}800 \times T}{D}$ for the Board of Trade, or $\frac{8{,}000 \times T}{D}$ for Lloyd's, which shows its empirical nature.

Valuable experiments on Adamson's rings were conducted by Messrs. Hall, Russell & Co., of Aberdeen, in 1882, and by Messrs. J. Howden & Co., of Glasgow, in 1887.

In Messrs. Hall, Russell & Co.'s experiment (see Fig. 1) only one of the rings was collapsed, and after it had been temporarily strengthened the experiment was again proceeded with. Unfortunately when the pressure was raised about 40 pounds above that at which the ring had collapsed, the outer cylinder burst and the experiment was abandoned. The coefficient given by the collapsed ring was:

$$\frac{P \times D}{T} = 64{,}213,$$

the diameter being taken over the plain part, and on careful measurement the collapsed ring was found to be of less thickness than any of the others.

Messrs. J. Howden & Co. experimented upon a furnace formed with four rings, each 23 inches long, and after each ring collapsed it was strengthened and the pressure still further increased

MESSRS. HALL, RUSSELL & CO'S. EXPERIMENTS.
SECTION OF FURNACE FORMED WITH ADAMSON'S RINGS.

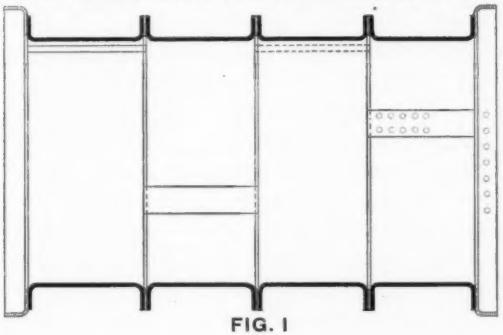


FIG. 1

MESSRS. JAMES HOWDEN & CO'S. EXPERIMENTS
SECTION OF ADAMSON'S RINGS.

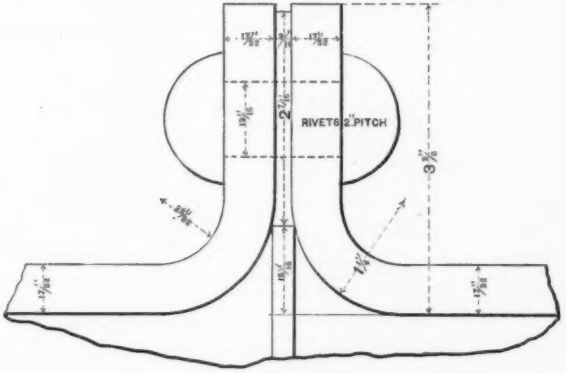


FIG. 2

SECTION OF FOX'S PATENT FURNACE
IN TESTING APPARATUS.

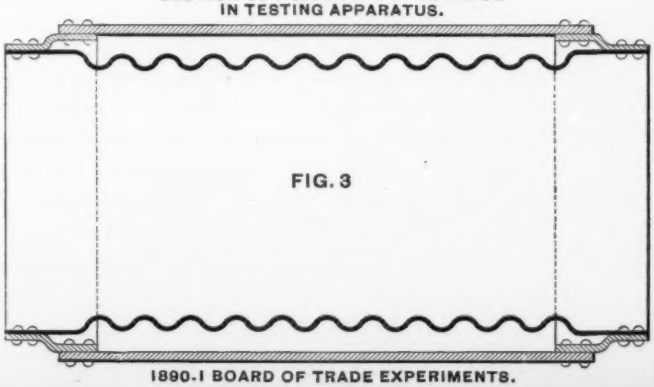


FIG. 3

until all the rings had been collapsed. Detailed particulars are given in Table I, the collapsing coefficient being 64,240, which agrees remarably well with the results obtained by Messrs. Hall, Russell & Co., although it should be noted that the length of the rings in Messrs. Howden's experiment was 23 inches, and that of Messrs. Hall, Russell & Co. only 19 inches. A section of the flanges of the rings is shown in Fig. 2.

There is no actual record available of the mechanical tests of the steel from which these furnaces were made, but they are understood to have been of the ordinary boiler quality manufactured at that time, and from 26 to 30 tons tensile strength.

The rule adopted by Lloyd's for furnaces made with Adamson's rings, with steel from 26 to 30 tons tensile, is:

Working pressure in pounds per square inch

$$= \frac{1{,}000 \times (T - 2)}{D},$$

where T = thickness of plate in sixteenths of an inch.

D = diameter of plain part of furnace.

and by the Board of Trade for steel from 26 to 30 tons:

$$\frac{8{,}800 \times T}{3 \times D}\left(5 - \frac{L + 12}{67.5 \times T}\right) = \text{working pressure in pounds per square inch.}$$

where T = thickness of plate in inches.

L = distance between centers of flanges in inches.

D = diameter of furnace in inches outside the flats.

Table II gives the tests of the Fox furnace in 1882 and 1891.

In November, 1882, the first test was made upon a single tube of very mild steel, having a tensile strength of 22.7 tons per square inch. Until 1890, all the furnaces were made of this low tensile steel, and the results obtained as compared with the furnaces made of the higher tensile, show, in the writer's opinion, that the step, however necessary from a commercial and competitive point of view, has not been successful in practice.

This test furnace was 6 feet 9 inches in length by 3 feet $1\frac{7}{8}$ inches outside diameter, and had thirteen corrugations 2 inches deep from outside to inside, pitched 6 inches from center to cen-

ter. Each furnace was enclosed in a steel cylinder, specially made for the purpose, and connected to it by rivets, as shown in Fig. 3. Two Dewrance's gauges were used to determine the pressure in the space between the furnace tube and the testing cylinder, the space being filled with water, and the requisite pressure applied by means of a force pump. Before any pressure was applied the furnace was carefully gauged, and is reported to have been truly cylindrical throughout its whole length.

In considering the results of this test, however, it should be remembered that the manufacture was practically in its infancy, and not nearly the same care was then exercised either in the manufacture or the method of recording the tests; consequently it may be safely assumed that, if a Fox furnace made of the same steel were tested now, the results would be better. For example, measurements were only taken at three points throughout the length of the flue, and it is recorded to have been absolutely cylindrical, although no other furnace experimented upon since has been so accurate. At each increment of 100 lbs. per square inch the furnace was gauged so that under various pressures the limit of elasticity and the ultimate strength to resist collapse should be ascertained. Up to 500 lbs. per square inch no alteration in the cylindrical form of the furnace was perceptible; at 800 lbs. it had altered its form to the extent of $\frac{1}{32}$ of an inch, but on the pressure being released it entirely recovered its original shape; permanent set could be distinctly measured at 850 lbs., and at a pressure of 900 it slowly collapsed, altering its form more readily than could be followed up by the pump. After collapse twelve holes were drilled in the furnace, and the thickness was found to vary from .51 to .53 of an inch.

It should be noted that until recently the Board of Trade took the mean diameter, viz., the mean between the least internal and greatest external. Lloyd's, however, take the greatest external, but, for comparison of results given in this paper, the diameter taken for the strength coefficient will be the mean diameter for furnaces of the Fox, Morison, and Farnley types, and the least external diameter for the Adamson, Holmes, and Purves designs.

The collapsing coefficient given by the test is:

$$\frac{900 \times 35.875}{52} = 62{,}091,$$

and the formula adopted by the Board of Trade for steel under 26 tons tensile, as a result of the experiment, was:

$$\frac{12{,}500 \times \text{thickness in inches}}{\text{mean diameter in inches}} = \text{working pressure in pounds}$$

per square inch, and by Lloyd's for the same limit:

$$\frac{1{,}000\ (T-2)}{D} = \text{working pressure in pounds}$$

per square inch, where T is the thickness of the plate in sixteenths of an inch and D the greatest diameter of the furnace in inches.

The Board of Trade formula gives a working pressure of 181.2 for the above furnace, and Lloyd's a working pressure of 166.8, so that on a basis of 22.7 tensile the factor of safety by Board of Trade rules was 4.97, and by Lloyd's 5.39. These factors are important, as they were the first granted for a modern furnace, and the experiment forms the basis of all subsequent tests, although, now, six furnaces are demanded for testing to destruction instead of one, and the severity of the requirements and the exactness of the tests are apparently increasing with each series. A higher value was granted by the Board of Trade than by Lloyd's for this furnace, but that the former was a perfectly safe allowance has been fully proved by experience, and will be referred to subsequently. From November, 1882, until 1890 the Fox furnace was made of steel between 21 and 25 tons tensile, but in the latter year it became necessary, for commercial reasons, to adopt a harder steel with a view of obtaining a higher coefficient than that which had been in successful use during the previous eight years. Accordingly, six furnaces were prepared and tested between the 4th November, 1889, and the 11th February, 1890.

Full particulars of these furnaces are given in Table II, the pitch and depth of the corrugations were the same as in the first test, the radical difference being that the steel employed had a mean tensile strength of 29.07 tons as against 22.7 for the first test. The second furnace of this series is said to have collapsed

at the 6 inch plain part where the thickness was only .331 of an inch, the mean thickness being .376. For this reason it has been proposed to abandon this test and take a mean of the other five, but as these furnaces were presumably made for purposes of experiment at least as much care would be exercised in their manufacture as in those intended for sale, and therefore little injustice would be done by including this test with the others. Of the six furnaces tested four collapsed on the plain part at the ends, one collapsed on the end corrugation, and the other on the second corrugation.

The mean collapsing coefficient from the six tests is 80,931, the difference between 62,091 and 80,931 being principally due to the increased strength of the steel, as may be seen by simple proportion:

$$\frac{62{,}091 \times 29.07}{22.7} = 79{,}514,$$

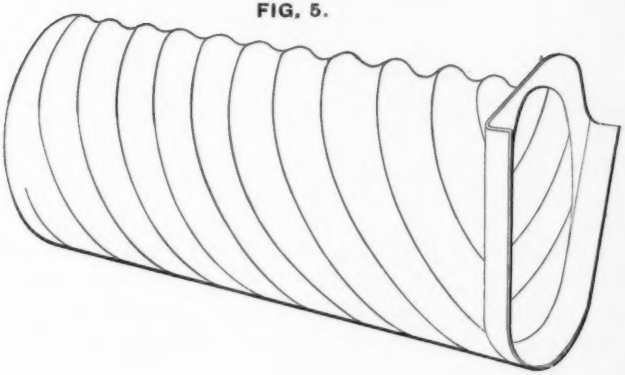
and when it is remembered that these flues were made with a degree of accuracy resulting from an experience of eight years, and allowing a margin for the imperfections of the 1882 test, it is evident that the strength of corrugated furnaces under cold water pressure varies directly as the tensile strength of the steel from which they are made.

Five tests of the Farnley flue were made under the Board of Trade in May, 1888, the results of which are given in Table III. The record of these tests is not quite so complete as many of the others; the steel being, it is understood, of ordinary boiler quality is assumed to have had a tensile strength of 28 tons per square inch, and the mean collapsing co-efficient when reduced to 27 tons has a value of 55,122, which is lower than that obtained with Adamson's rings. It should, however, be noted that the Farnley Iron Company have now improved their method of manufacture, and can make furnaces much more truly round and therefore much stronger than when these experiments were made. A perspective view of the Farnley furnace is given in Fig. 5.

Table IV gives the test results of the Holmes design (see Figs. 6 and 7). Two of these flues collapsed at the plain part between two corrugations, and in the other case a corrugation and the

FARNLEY PATENT FURNACE.

FIG. 5.



HOLMES' PATENT FURNACE.

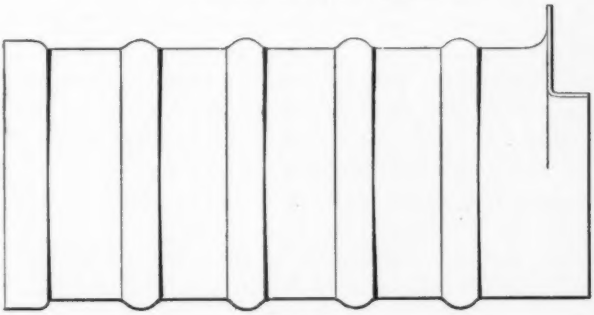


FIG. 6.

SECTION OF PURVES' PATENT FURNACE.
IN TESTING APPARATUS.

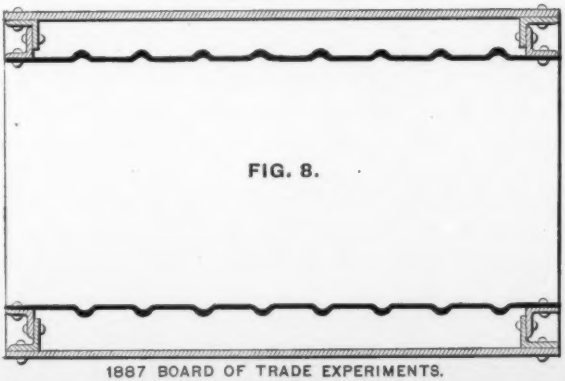


FIG. 8.

adjacent plain part collapsed. The mean collapsing coefficient given by the three tests is 60,995, and the mean tensile strength of the steel was 27.2 tons per square inch.

Messrs. John Brown & Co., of Sheffield, made the first test with the Purves flue in March, 1887. Two flues were tested, each nominally $\frac{9}{16}$ inch thick and 6 feet $6\frac{3}{8}$ inches long between the centers of the inner row of rivets by which it was secured to the angle iron bars. There were seven ribs with a pitch of 9 inches, and the two end ribs were $8\frac{11}{16}$ and $15\frac{11}{16}$ inches respectively from the centers of the inner row of rivets. The end flats were unusually long, and were, in fact, the weakest part of the flue. Each rib projected 1 inch above the level of the plate on the outside; the plate was grooved on the inside to a depth nearly equal to the thickness of the plate. The greatest deviation of the first tube from the circular form was $\frac{7}{32}$ of an inch, and of the second tube $\frac{11}{32}$ of an inch. Collapse took place with the first tube at 740 pounds on the $15\frac{11}{16}$ plain part about 18 inches from the weld, the second tube collapsing at 760 pounds, also at the long flat in the front end, and about 24 inches from the weld. Measurements were taken at twelve test holes in each case, and the thickness was found to be .563 in the first and .546 in the second.

Taking the external diameter of the flats as being the diameter acted upon by the water, the collapsing coefficients are:

$$\frac{740 \times 38.61}{568} = 50{,}302;$$

$$\frac{760 \times 38.685}{546} = 53{,}837;$$

or a mean of 52,074. The mean tensile strength of the material was, unannealed, 26.9 tons, and as this result was much inferior to that obtained by Messrs. Hall, Russell & Co., and Jas. Howden & Co., with Adamson rings, and as these flues were somewhat roughly made, Messrs. John Brown & Co. arranged another series of tests.

Of this second series, two furnaces were nominally $\frac{5}{16}$, two $\frac{7}{16}$, and two $\frac{9}{16}$ thick. Each of the $\frac{7}{16}$ and $\frac{9}{16}$ flues were made from a plate which had been rolled to the form and thickness required,

but the $\frac{5}{16}$ tubes were made from plates rolled similar to the $\frac{9}{16}$, and afterwards planed on the ribbed side to the required thickness. Each tube was 6 feet $7\frac{1}{2}$ inches in length between the centers of the inner rows of rivets, and there were eight stiffering ribs pitched 9 inches apart. The centers of the two outer ribs were $6\frac{3}{4}$ and $9\frac{3}{4}$ inches respectively from the centers of the inner rows of rivets in the end rings; the ribs projected about $1\frac{5}{16}$ inches above the outer surface of the tubes, and were grooved to a depth of about $\frac{3}{4}$ inch, as shown in Figs. 9, 10, 11, 12, the various results of the tests being given in Table V. A section of the furnace in position in the test tube is also given in Fig. 8.

A point to be noted in this second series of tests of the Purves flue is the rapid reduction of the collapsing coefficients as the thickness of the flue increases; thus, the mean collapsing coefficient for $\frac{5}{16}$ inch, $\frac{7}{16}$ inch, and $\frac{9}{16}$ inch were respectively 81,980, 58,327, and 55,435. The chief cause of this reduction in strengh is that the amount of metal in the strengthening ribs is practically a constant quantity, and therefore bears a less proportion to the total in thick flues than in thin.

Comparing the results obtained from the $\frac{9}{16}$ furnace in the second series of tests with the results obtained from the same thickness in the first set, the collapsing coefficients are respectively 55,435 and 52,074. The slight gain, amounting to about $6\frac{1}{2}$ per cent., may be due to the increased tensile strength of the steel (27.9 against 26.9) to the shorter flats at the ends or to the increased depth of the strengthening ribs. These causes are sufficient to account for a greater increase of strength than was actually obtained, and it is probable that had the $\frac{7}{16}$ and $\frac{9}{16}$ furnaces been more satisfactory in form, a higher collapsing coefficient would have been obtained.

As it is very seldom that $\frac{5}{16}$ furnace flues are used in practice, the results of this thickness being so much in excess of the others should be neglected, and the mean of the $\frac{7}{16}$ and $\frac{9}{16}$ taken as representing the strength of flues in actual use for marine boilers. The mean of these four tests is 56,881, which is still considerably under the coefficient obtained with Adamson's rings.

BOARD OF TRADE TESTS. 1897.
SECTION OF PURVES' PATENT FURNACE.

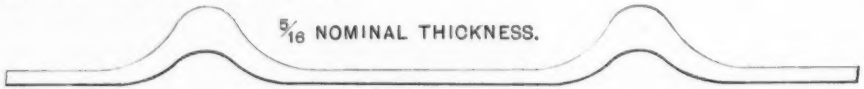


FIG. 9.

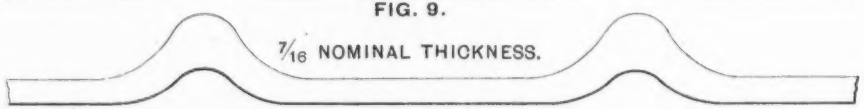


FIG. 10.

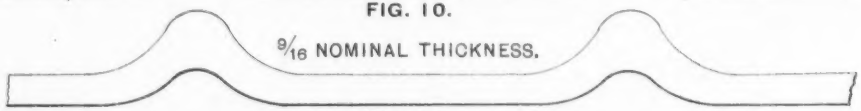


FIG. 11.

SECTIONS OF PURVES' & HOLMES' FURNACES.

FIG. 12.

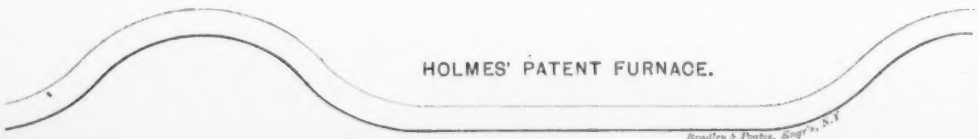


FIG. 7.

SECTIONS OF FOX'S & MORISON'S FURNACES.

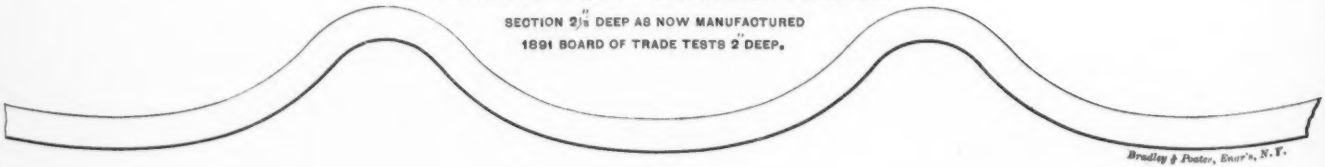
FOX'S PATENT FURNACE.

SECTION AS NOW MANUFACTURED
AND AS TESTED IN 1890-1

FIG. 4.

MORISON'S PATENT SUSPENSION FURNACE.

SECTION 2$\frac{11}{16}$ DEEP AS NOW MANUFACTURED
1891 BOARD OF TRADE TESTS 2" DEEP.



Bradley & Foster, Engr's, N.Y.

FIG. 14.

Messrs. John Brown & Co., being still dissatisfied, prepared another set of furnaces, which were tested in the same manner in 1889 by the Board of Trade only, and one furnace which was tested by Lloyd's in December, 1890. The results of these tests were much better, and are given in Table VI. The mean tensile strength of the steel used in the manufacture of these furnaces is given at 28.2 tons per square inch.

In September and November, 1891, experiments were made by the Leeds Forge Company upon six of Morison's suspension furnaces, the results of which are given in Table VII. These furnaces were made of steel having a mean tensile strength of 27.17 tons per square inch, and, as will be seen from the table, the results obtained were remarkably uniform, and may be considered absolutely reliable, as the severity of the Board of Trade requirements, the strict supervision of the entire process of manufacture, including the annealing of the furnaces prior to being riveted in the test tubes, were all carried out with a degree of exactness never exceeded. Of the six furnaces tested, two collapsed on the end flats, and the remaining four on the corrugations; the mean length of flat was $4\frac{3}{4}$ inches, and, as it is not necessary that the corrugations should be alternated in a boiler, no longer flat than this need ever be used in practice.

The average collapsing coefficient given by the six furnaces is 82,185, which is slightly higher than that given by the Fox flues, which were made of 29.07-ton steel. Reducing both to a basis of 27 tons, we have—

Morison's collapsing coefficient, . .	81,641
Fox's, ...	75,153

showing Morison's tests to have given a result $8\frac{1}{2}$ per cent. better than Fox's. The depth of the corrugations for these test furnaces was 2 inches, but as it was considered necessary that the body of a furnace should be the strongest part, the rolls were cut $\frac{1}{8}$ deeper, so the furnaces as since made are stronger than those tested. Fig. 13 shows Morison's furnace and test tube in section, and Fig. 14 gives a full-size section of the corrugations.

Diagram No. 18 shows the collapsing coefficients of the fur-

naces mentioned in this paper, the coefficients being calculated by the formula:

$$\frac{P \times D}{T} = \text{collapsing coefficient.}$$

The diameter D being the mean diameter in the cases of the Fox, Farnley and Morison furnaces, and the least external diameter for the Purves, Holmes and Adamson. The coefficients so found have been brought to the common basis of steel of 27 tons tensile strength so that they may be directly compared. As two furnaces of each thickness were tested in most cases, the mean has been taken and the points in the diagram show the position of these means.

The abscissæ show the thickness of the test furnaces and the normals represent the collapsing coefficients. The furnaces in practical use being generally of greater thickness than $\frac{7}{16}$, the furnaces under this thickness are shown in the diagram by a dotted line.

The tests of the suspension furnace are remarkably uniform, being represented by an almost horizontal line, thus indicating that the strength of each thickness of furnace is exactly proportional to the thickness of the plate. The low position of the $\frac{5}{16}$ Fox furnace may be ascribed to the fact that it was thin at one end, while the drooping line of the Purves tests shows the influence of the strengthening rib which is greater in thin furnaces than in thicker ones.

It would seem that the Farnley and Holmes tests had not been so accurately carried out as the others, both as regards the manufacture of the furnaces or the method of testing, or the tests would probably have been more uniform.

FLATS ON FURNACE ENDS.

The test value of modern marine furnaces depends to an extent on the length of the flat parts at the ends. Fig. 23 gives the results of those test furnaces which collapsed at the end flats, the collapsing coefficients having been calculated by taking the external diameter.

Where $\frac{P \times D}{T} = \text{collapsing coefficient.}$

DIAGRAM OF STRENGTH COEFFICIENTS CALCULATED ON THE MEAN DIAMETER OF THE MORISON, FOX, AND FARNLEY FURNACES.
AND THE DIAMETER OVER FLATS OF THE PURVES, HOLMES, AND ADAMSON FURNACES,
AND REDUCED TO A BASIS OF 27 TONS PER SQUARE INCH.

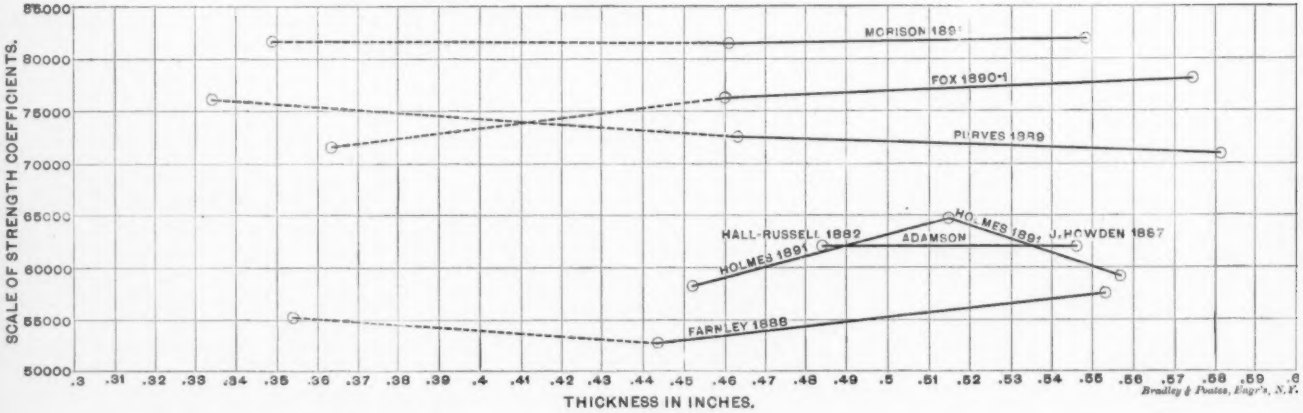


FIG. 18.

STRENGTHS OF FLATS AT FURNACE ENDS.

CALCULATED ON A BASIS OF STEEL HAVING A COMMON TENSILE STRENGTH OF 27 TONS PER SQ.IN.

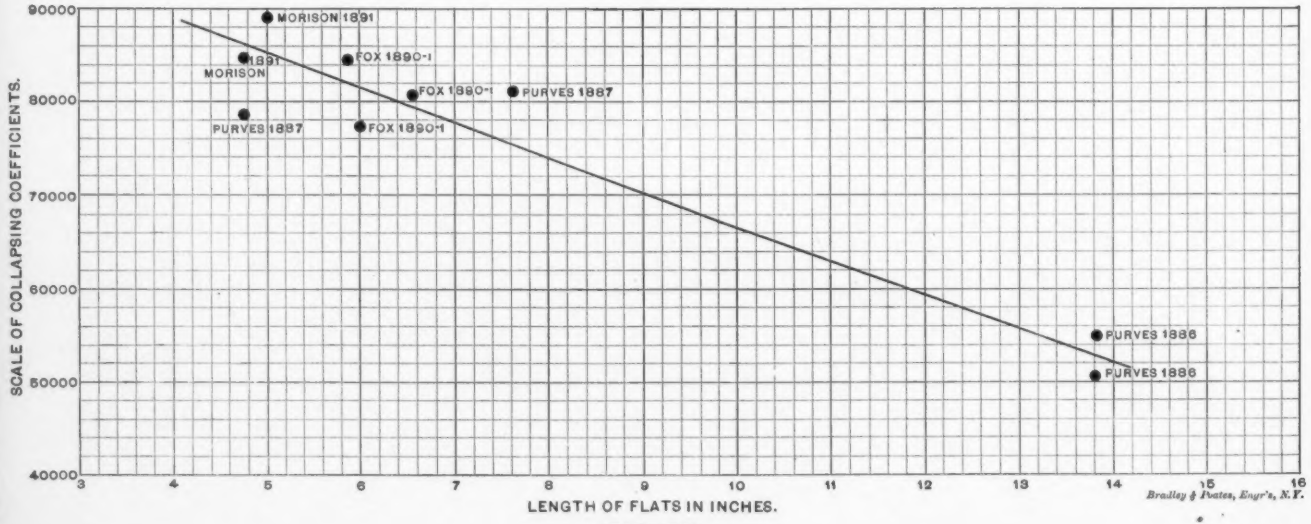


FIG. 23.

P = pressure per square inch at which collapse took place.
D = external diameter of the flat.
T = thickness of flat in inches.

In each case the collapsing coefficient has been reduced to a basis of steel having a tensile strength per square inch of 27 tons. The diagram includes all the Fox, Purves and Morison test furnaces which collapsed on the flat, the diagonal line representing the mean result.

In the Morison furnace no flat need exceed $4\frac{3}{4}$ inches, and the Board of Trade have fixed the limit at 5 inches, or 1 inch less than in the Fox, the explanation of this difference being that the design of the former is more suitable for short flats than the latter. By the aid of the diagram the collapsing coefficient of any length of flat can be found with any furnace of the modern types. Neglecting the irregularities of the tests, which may be due to one furnace being more nearly circular than another, it is seen that the strength of an end flat bears an inverse ratio to its length. The weakest flat is at the top back end over the bridge, but this in practice can be reduced to a minimum. Flats at the front end and the bottom back end are not exposed to such a high temperature, consequently the end plates in a furnace when in use are probably stronger than that part of the body of a furnace exposed to the greatest heat.

It has lately been the custom when the flat at the back bottom end exceeds the regulation limits to roll in a corrugation half way round in a Fox furnace and hammer out a rib half way round in a Purves design, but unless the flat is very long this appears unnecessary, as deformation or collapse does not generally take place in the ashpit except when the furnace is too rigidly stayed.

FACTORS OF SAFETY.

The furnace manufacture for the mercantile marine of this country is under the absolute control of the Board of Trade and Lloyd's Registry, as not only do these authorities determine the limits of the tensile strength of the steel to be used, but by formulating rules for the strength of furnaces they also control the

weight, and consequently the commercial value of every type of furnace used in marine boilers.

The importance of the furnace manufacture may be estimated from the fact that the Leeds Forge Company have made over 30,000 corrugated furnaces, while Messrs. John Brown & Co., of Sheffield, have made 10,000 of the Purves design. Until recent years, or since the general adoption of triple-expansion engines, the types of special furnaces were but few; the demand for improvement, however, has resulted in other designs, and the increasing severity and accuracy of the Board of Trade tests may be explained by the desire to give each design proportionate value. A tabulated report of the tests of the Suspension furnace is appended, which illustrates the highly expensive and thorough nature of the experiments. Table VIII gives the apparent factors of safety allowed by the Board of Trade and Lloyd's for the various furnaces mentioned in this paper; it also gives the factors reduced to a common basis of steel of 27 tons tensile strength per sq. inch. The factors of safety have been formed by dividing the actual collapsing pressures under cold-water test by the working pressures allowed by the Board of Trade and Lloyd's rules.

$$\text{Factor of safety} = \frac{\text{actual collapsing pressure}}{\text{pressure given by formula}}$$

The fluctations of the factors of safety required at different times for the same furnace, and at the same time for different furnaces, are shown by Fig. 19. In 1891 the Board of Trade required a factor for the Fox furnace of 5.58, and in 1892 it is 5.44. For the Purves furnace the factor in 1888 was 4.87, and in 1890 it was 5.05, whilst the Morison furnace has a provisional factor of 5.91.

Lloyd's factors also vary considerably. In 1888 the Fox factor was 5.35, and in 1892 it was 5.24. The factor of the Purves furnace in 1888 was only 4.21; in 1889, 4.21, and in 1890, 4 85. The Holmes furnace requires 5.34; the Farnley, 5.39, and the Suspension, 5.58.

These furnaces being all of high tensile steel, cannot be directly compared with the first of the Fox furnace in 1882. The larger factors now demanded seem to indicate a decrease of con-

XU

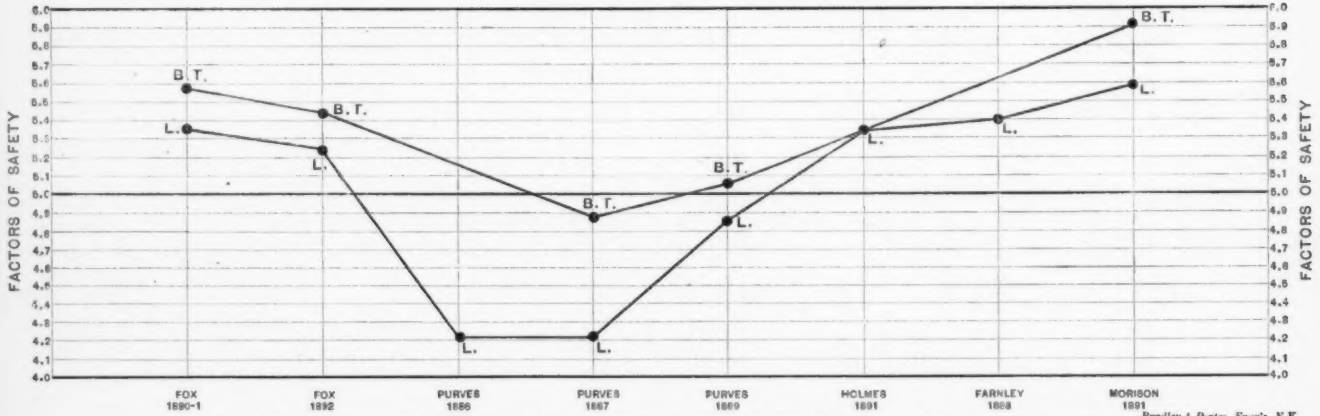
T

Date of Test.	Description of Furnace.	Board of Trade or Lloyd's.	Diameter
Nov., 1882.....	Fox.....	Board Trade	
	"	Lloyd's.....	over corr
Nov. 4, 1890, and Feb. 11, 1891.	"	Board Trade	
	"	" "	
	"	Lloyd's.....	over corr
	"	"	" "
Same test as above, but taking the Board of Trade and Lloyd's rules for 1892.	"	Board Trade	least ext
	"	" "	"
	"	Lloyd's.....	over corr
	"	"	"
Mar. 12, 1886....	Purves...	Lloyd's	{ least extern { diam. over
	" ...	"	
1887	" ...	Board Trade	"
	" ...	" "	"
	" ...	Lloyd's.....	"
	" ...	"	"
1889	" ...	Board Trade	"
	" ...	" "	"
	" ...	Lloyd's.....	"
	" ...	"	"
1891	Holmes..	Lloyd's.....	{ least extern { diam. over
	" ..	"	
May, 1888.....	Farnley..	Lloyd's.....	over corr
	" ..	"	"
Sept. 25 and 26, 1891, with 1892 coefficients.	Morison..	Board Trade	least ext
	" ..	" "	"
	" ..	Lloyd's	over corr
	" ..	"	"

TABLE VIII.—FACTORS OF SAFETY.

Board of Trade Lloyd's.	Diameter of Furnace.	Mean Thickness.	Collapsing coefficients, taking the mean diameter for Fox, Farnley, and Morison furnaces, and least external diameter for Purves' and Holmes' furnace.		Collapsing Pressure.	Coefficient Allowed.	Working Pressure.	Factor of Safety.	Ultimate Tensile Strength of Steel.	Factor of Safety reduced to Steel of 27 tons.
			Actual.	Reduced to steel of 27 tons tensile.						
d Trade	mean 35.875	.52	62,091	73,852	900	12,500	181.2	4 97	22.7	
d's.....	over corrgtns. 37 875	.52	62,091	73,852	900	1,000	166.8	5.39	22.7	
d Trade	mean 33.593	.574	81,934	77,485	1,400	14,000	239.2	5.85		
"	" 34.937	.575	85,671	78,650	1,410	14,000	230.4	6.12		
							mean	5.98	28 98	5.58
d's.....	over corrgtns. 35.593	.574	81,934	77,485	1,400	1,234	249.1	5.62		
.....	" " 36.937	.575	85,671	78,650	1,410	1,234	240.5	5 86		
							mean	5.74	28.98	5.35
d Trade	least external 32.741	.574	81,934	77,485	1,400	14,000	245.4	5 70		
"	" " 34.087	.575	85,671	78,650	1,410	14,000	236.1	5.97		
							mean	5.84	28.98	5.44
d's.....	over corrgtns. 35 593	.574	81,934	77,485	1,400	1,259	254 1	5.51		
.....	" " 36.937	.575	85,671	78,650	1,410	1,259	245.4	5.74		
							mean	5 63	28.98	5.24
d's.	{least external or diam. over flats.} 38 61	.568	50,302	49,032	740	1,000	183.6	4.03		
.....	{least external or diam. over flats.} 38.685	.546	53,847	54,656	760	1,000	174.1	4 36		
							mean	4 20	26.9	4.21
d Trade	" " 38.847	.611	55,632	54,422	875	11,000	173.0	5.06		
"	" " 38.788	.613	55,239	52,701	873	11,000	173.8	5.02		
							mean	5.04	27.95	4.87
d's.....	" " 38.847	.611	55,632	54,422	875	1,000	200.2	4.37		
.....	" " 38.788	.613	55.239	52,701	873	1,000	201 3	4.34		
							mean	4.35	27.95	4.21
d Trade	" " 38.63	.585	70,326	66,160	1,063	14,000	212.0	5.02		
"	" " 38.65	.578	76,564	75,446	1,145	14,000	209.4	5.47		
							mean	5.24	28.05	5.05
d's.....	" " 38.63	.585	70,326	66,160	1,065	1,160	221.0	4.82		
.....	" " 38.65	.578	76,564	75.446	1,145	1,160	217.5	5.27		
							mean	5.04	28 05	4.85
d's.....	{least external or diam. over flats.} 35.37	.515	65,245	64,528	950	945	166.7	5.70		
.....	{least external or diam. over flats.} 35.5	.557	58,635	59,072	920	945	184.0	5.00		
							mean	5.35	27 05	5.34
d's.....	over corrgtns. 41.54	.559	58,601	56,508	835	912	152.5	5.48		
.....	" " 41.43	.548	60,697	58,529	850	912	149.0	5.70		
							mean	5.59	Assmd. 28	5.39
d Trade	least external 33.779	.530	85,159	83,550	1,300	13,500	211 7	6.14		
"	" " 33.212	.567	80,537	80,269	1,340	13,500	230.5	5.81		
							mean	5.98	27.305	5.91
d's.	over corrgtns. 36.719	.530	85,159	83,550	1,300	1,259	222.2	5.85		
.....	" " 36.078	.567	80,537	80,269	1,340	1,259	246.8	5.43		
							mean	5 64	27.305	5.58

DIAGRAM SHOWING THE FACTORS OF SAFETY ALLOWED BY THE BOARD OF TRADE AND LLOYDS' REGISTRY FOR VARIOUS FORMS OF FURNACES, ON THE BASIS OF COLD WATER TESTS.



FACTORS CALCULATED ON STEEL HAVING A COMMON TENSILE STRENGTH OF 27 TONS PER SQ: IN:

Bradley & Yeates, Engr's, N.Y.

FIG. 19.

fidence in steel of the high tensile strength, such as has lately been used. The Board of Trade have now adopted the standard dimension of the least external diameter as a basis for their formula for all furnaces, and, as the least internal diameter is a measure of the effective width of grate, this dimension seems very suitable and convenient. Lloyd's formula require in the case of the Fox, Farnley and Morison furnaces the greatest external diameter, and for the Purves, Holmes and Adamson the least external diameter. This difference causes a difficulty in directly comparing the constants allowed for different furnaces, and a standard dimension of the least external diameter would be a great improvement.

MATERIAL.

It is impossible to over-estimate the value of determining the quality of steel most suitable for the manufacture of boiler furnace tubes of modern high pressure boilers, as suitability of material is the first essential to practical success. On the introduction of the corrugated furnace in 1882, the manufacturers considered that a steel of low tensile strength and high elongation was the best for the general requirements of a furnace, and the fact that, at the Leeds Forge alone, 28,000 furnaces have been made of steel under 26 tons tensile, and that in 1888 the Engineer-in-Chief of the Board of Trade should have written the following in his work on "Boiler Construction," is a sufficient proof that their views were correct.

Mr. Traill says: "Up to this time, 1888, it is not known that any serious casualty has resulted from the collapse of a corrugated furnace, although some have come down, but, when this has been the case, the plates, it is believed, have been overheated, due to dirt or other easily assigned cause not attributable to defects in the furnace, and such deformations as have taken place would have been of a much more serious nature had the furnace been plain instead of corrugated." Mr. Traill also foreshadows a modification of the Fox design, such as the suspension furnace, as he says, "there is still a good deal of vitality in the corrugated furnace, and possibly its special advantages

may yet be improved upon, and so give it new life to compete successfully with any that may come into the market against it."

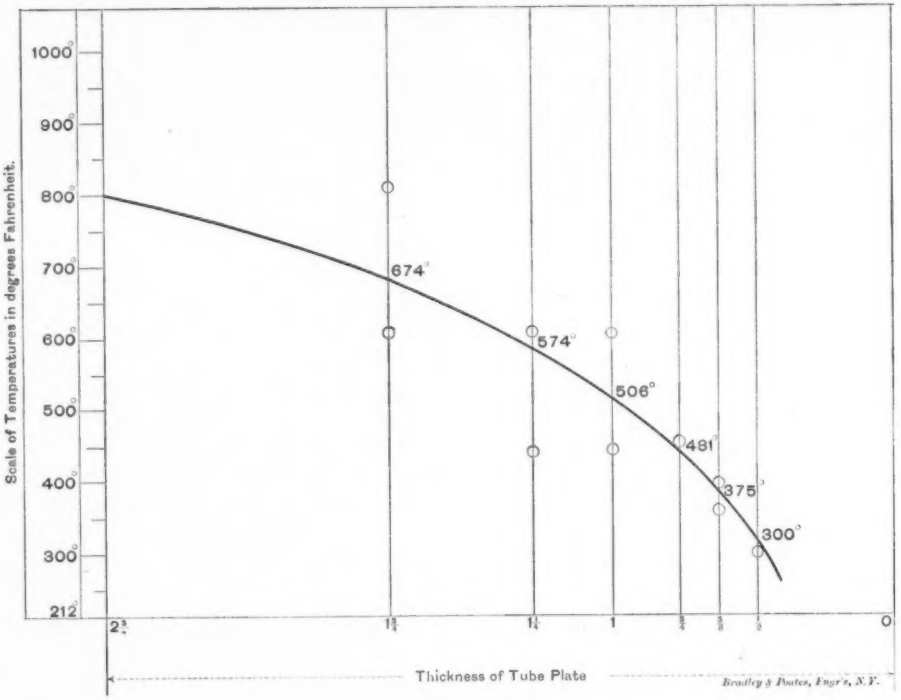
The Board of Trade regulations for furnace steel are that it should not be less than 26 tons tensile strength and should not exceed 30 tons, and recognize 28 tons per square inch with an elongation of not less than 20 per cent. in 10 inches as a good tensile strength such as is suitable for furnaces. Lloyd's also recognize steel of from 26 to 30 tons as suitable for furnaces. The English Admiralty, whose experience with corrugated furnaces of low tensile strength has been remarkably successful, now specify that the tensile strength of steel for furnaces must not exceed 25 tons with an elongation of not less than 27 per cent. in 8 inches. All Continental governments also use steel under 26 tons for furnaces, and the United States Government requires the tensile strength to be from 50,000 to 60,000 pounds or 22.3 to 26.8 tons per square inch.

The tensile strength of the Purves flue has always been from 26 to 30 tons for the mercantile marine; consequently, from a commercial point of view, the Leeds Forge Company were compelled to raise their standard tensile strength, and since 1890 practically all furnaces for the mercantile marine have been made of steel having an average tensile strength of 28 tons, or 4 tons higher than the standard adopted by the Leeds Forge Company prior to that date, and 4 tons higher than the requirements of the English and foreign governments.

The all important question is, therefore, have furnaces made from steel of from 26 to 30 tons tensile been as successful as those made from 23 to 26 tons? The writer has lately consulted with many eminent engineers on this important subject, and the general opinion undoubtedly is that we may pay too dearly for the lighter furnaces of the harder material, and the fact that the percentage of failures is now greater than ever before has caused a feeling in favor of softer material, or, in other words, that steel having a tensile strength not exceeding 26 tons with an elongation of 25 per cent. in 8 inches is more suitable for furnaces than steel having a tensile strength not exceeding 30 tons.

FIG. 24.

DR. KIRK'S EXPERIMENTS WITH DIFFERENT THICKNESSES OF TUBE PLATES.



The advantages for the lower standard are, in the writer's opinion:

It is better able to stand the rough usuage of an ordinary boiler shop.

It is less liable to develop small cracks in manufacture which may pass unnoticed and afterwards extend.

It is more ductile, and will stand the continued expansion and contraction incidental to a steam boiler better than steel of high tensile strength.

It will remain longer ductile than the harder steel.

It will also be less liable to crack at the back ends.

In fact, the only point which is apparently in favor of the steel of high tensile strength is that the first cost of the furnace is slightly less, although the result is uncertainty of material and the probable shortened life of the furnace.

TEMPERATURE.

The influence of temperature upon the strength and ductility of mild steel does not seem to have had the attention that the subject merits. There is considerable uncertainty as to the exact temperature of the material of a furnace under actual working condition; but the experiments made by the late Dr. A. C. Kirk have determined it with approximate accuracy. [See JOURNAL for August, 1892, Vol. IV, p. 507.]

Dr. Kirk aimed at finding the temperature of the fire side of tube plates of various thickness, the upper sides of which were covered with water, which was allowed to boil at atmospheric pressure. The results of the experiments are shown graphically in Fig. 24, the abscissæ representing thickness of tube plate and the normals the temperature of the fire side of the plate above 212° Fahr. A fair curve is drawn through the mean of the results.

With a clean plate 1 inch thick, the temperature of the fire side was found to exceed that of the water side by 294° F., while with $\frac{1}{2}$-inch plate the difference under the same conditions was only 88° F.

With a temperature on the water side of 380° F., correspond-

ing to a steam pressure of 180 pounds per square inch, the temperature on the fire side of an inch plate would be 670°, and with a ½-inch plate 468° F.

The diagram shows that the difference in temperature increases from 88° F., with a ½-inch plate, to 588° F., with a 2¾-inches plate.

It may be noted that the results are not so reliable for very thick plates, but within the limits of thickness used in practice they are sufficiently accurate.

The thickness of the plate of which a furnace is formed has a most material influence upon the temperature of the furnace, and, as in practice the thickness varies between the limits of ½ inch and 1 inch, it may be assumed that the temperature (when the furnaces are clean and have no scale) is between 468° and 670° F.

The influence of temperature upon the behavior of mild steel under tension is shown by a splendid set of experiments made with the United States testing machine in 1888, from which the diagrams, Figs. 20, 21 and 22, have been copied.

Briefly, the results shown by these experiments are—that the tensile strength of all steel decreases from zero F. to a temperature of 200° to 300° F.; this is followed by an increase, and the maximum tensile strength is reached at about 400° to 600° F.; beyond this latter temperature the strength decreases rapidly.

Although the ultimate tensile strength of the steel increases between 200° and 600° F., the elastic limit steadily decreases from zero F. upwards, and steel having an elastic limit of 35,000 pounds per square inch at zero has its elastic limit reduced to 20,000 pounds per square inch at 600° F.

It is also found that between zero and 550° F. all steels show decided yielding at loads below the elastic limit, and that after this yielding takes place it may go on even with slightly reduced loads. This is especially the case with mild steels, such as are used for the manufacture of furnaces.

Another point is, that the higher carbon steels reach the temperature of maximum strength more abruptly, and retain their

EXPERIMENTS ON STEEL, MADE BY THE UNITED STATES GOVERNMENT.

DIAGRAM OF TENSILE STRENGTH AT VARYING TEMPERATURES.

CHEMICAL COMPOSITION { CURVE A, CARBON, .09; MANGANESE, .11
" B, " .20 " .45

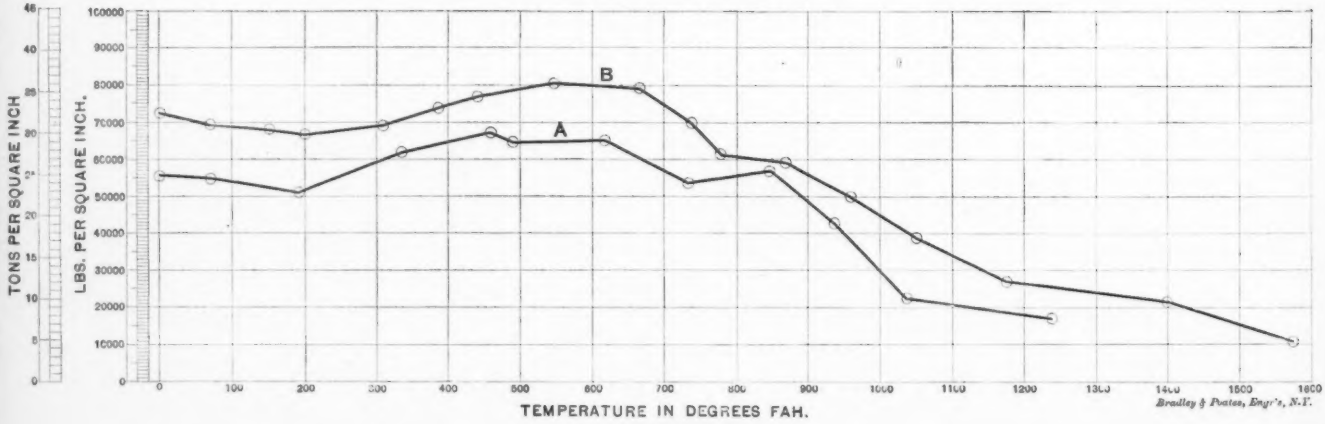


FIG. 20.

EXPERIMENTS ON STEEL, MADE BY THE UNITED STATES GOVERNMENT.

DIAGRAM OF ELASTIC LIMIT UNDER TENSION AT VARYING TEMPERATURES.

CHEMICAL COMPOSITION { CURVE A, CARBON, .09; MANGANESE, .11
" B, " .20 " .46

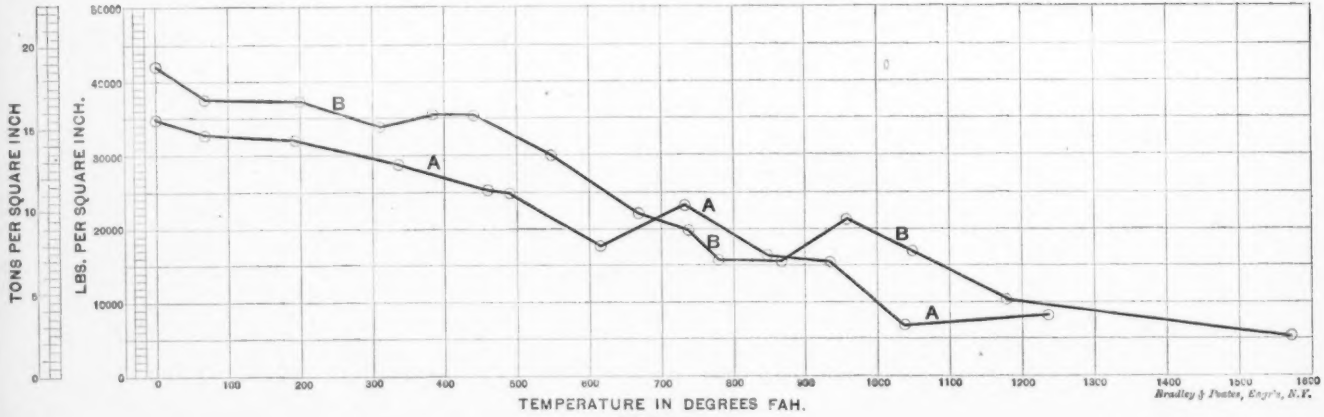


FIG. 21.

EXPERIMENTS ON STEEL, MADE BY THE UNITED STATES GOVERNMENT.

DIAGRAM OF CONTRACTION OF AREA UNDER TENSION AT VARYING TEMPERATURES.

CHEMICAL COMPOSITION { CURVE A, CARBON, .09; MANGANESE, .11.
" B, " .20; " .45.

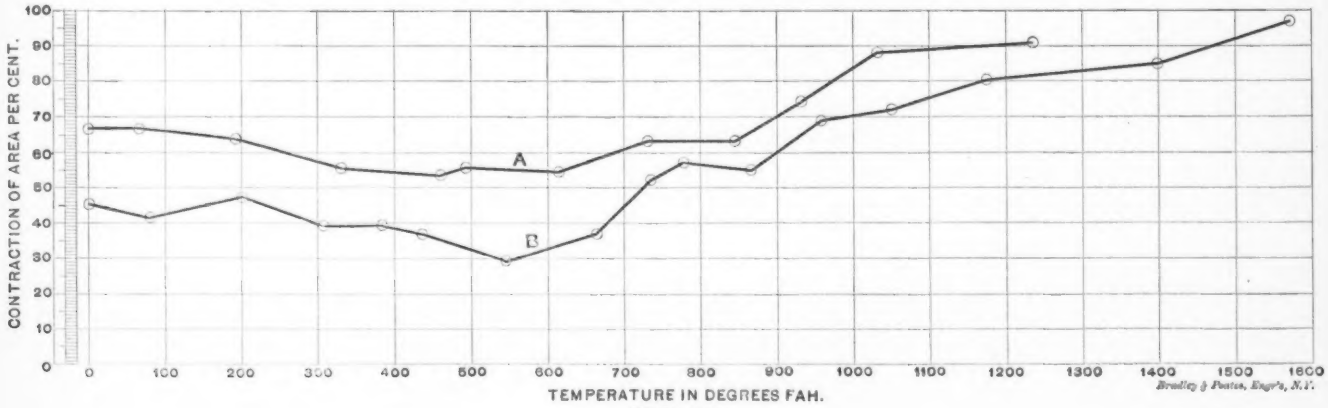


FIG. 22.

highest strength over a less range of temperature than steels having a low percentage of carbon.

The ductility of steel also varies with the temperature; this is shown by Fig. 22, which represents the contraction of area at the point of fracture for various temperatures. There is a tendency to fracture with less contraction of area up to a temperature of 400° to 600° F. In the vicinity of this range of temperature a number of the bars fractured obliquely across their stems instead of at right angles as at higher and lower temperatures. This behavior is significant, and appears to connect the tensile tests with the brittleness which is also observed in bending tests at these temperatures.

The influence of temperature upon steel may be summarized as follows:

With a thick furnace the temperature of the plate will be higher than with a thin one, and consequently the elastic limit will be lower, and there will be a liability to sag unless the factor of safety is very high in the first instance.

With thin furnaces, say about $\frac{1}{2}$-inch to $\frac{5}{8}$-inch thick and 180 pounds pressure of steam, the temperature of the inner surface of furnace plate, which is clean on the water side, is 470°, and consequently about the brittle temperature; it is therefore necessary to guard against this brittleness by employing steel of low tensile strength, as the lower carbon steels retain their maximum strength and ductility over a greater range of temperature than the higher carbon steels.

LONGITUDINAL ELASTICITY OF FURNACES.

The following particulars are extracted from a paper recently read in Germany by Herr Von Knaudt, of the firm of Schulz, Knaudt & Co., of Essen, on "Experiments Made to Determine the Longitudinal Elasticity of Various Designs of Furnaces." These experiments were evidently made at great cost, and their completeness and reliability form a most valuable addition to furnace literature. The experimental apparatus consisted of an hydraulic press (shown in Fig. 16), in which the furnace to be tested was placed horizontally between the body of the press

and the hydraulic ram, these two representing the ends of a boiler. The load was applied longitudinally in increments of 15 pounds, each increment on the pump representing 20.28 tons on the ram. On the completion of each increment the shortening of the furnace was measured, the pressure was taken off, and the length again measured in order to ascertain the permanent set, if any. At intervals the pressure was allowed to remain on for a time, then released, and again applied, but there was practically no difference in the results if no permanent set had taken place. The results of these experiments are shown on Fig. 17, the horizontal scale representing the load in tons applied to the ends of the furnace, and the vertical scale represents in decimals of an inch the compression of the furnace in a length of 36 inches. In the curves the full lines denote the results under pressure, and the dotted lines the results after the pressure was released. The furnaces experimented on were five of the corrugated type, one Adamson and one Purves ribbed. All these were made by Messrs. Schulz, Knaudt & Co., of Essen, with the exception of the Purves flue, which was manufactured by Sir John Brown & Co., Sheffield. The tensile strength of the steel for four of the corrugated furnaces, Nos. 1, 3, 4 and 5 on the diagram, and also the Adamson No. 6, was between 21 and 25.4 tons per square inch, with an elongation of 25 to 30 per cent., and the fifth corrugated, or No. 2, was 29.2 tons tensile strength and 28 per cent. elongation.

The tensile strength of the plate from which the Purves flue was made was not ascertained, but was probably between 26 and 30 tons. Full particulars of the dimensions of each furnace are given in Figure 16. For the comparison of the results take a uniform pressure of 100 tons applied at the end of the furnace. In the corrugated furnaces the compressions were .134, .112, .093, .075 inches per 36 inches in length, in the Adamson .021, and in the Purves .0071 inches. Further, the pressures required to shorten these furnaces $\frac{1}{32}$ of an inch (or .03125) per 36 inches in length, would be for Nos. 1, 2, 3 and 4, 19.27, 24.74, 30.73, and 41.84 tons respectively; No. 5, 92.18; No. 6, 145.22; whilst to compress No. 7 would take 300 tons.

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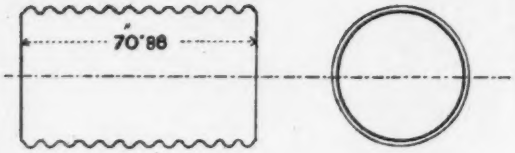
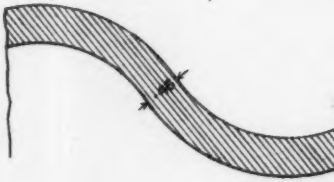


EXPERIMENT No. I.	
Greatest Outside Diameter - -	41·14
Least Inside Diameter - - -	37·28
Tensile Strength of Steel - -	28·5
Elongation of Steel - - - -	25%



EXPERIMEN
Greatest Outside Diam
Least Inside Diameter
Tensile Strength of St
Elongation of Steel -

EXPERIMENTS MADE BY HERR VON KNA

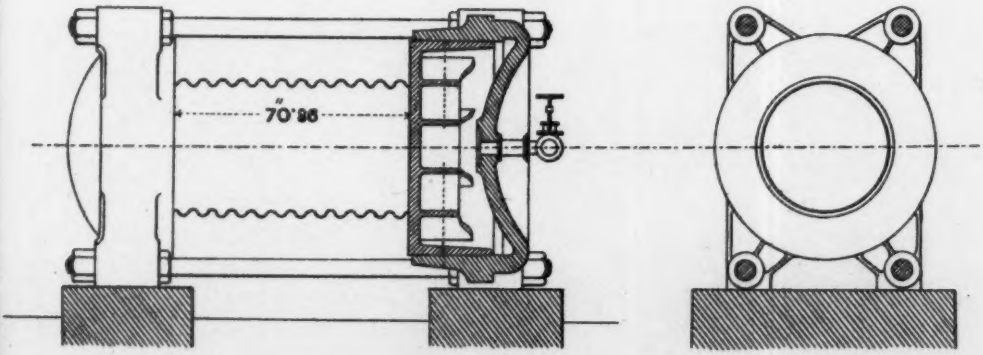
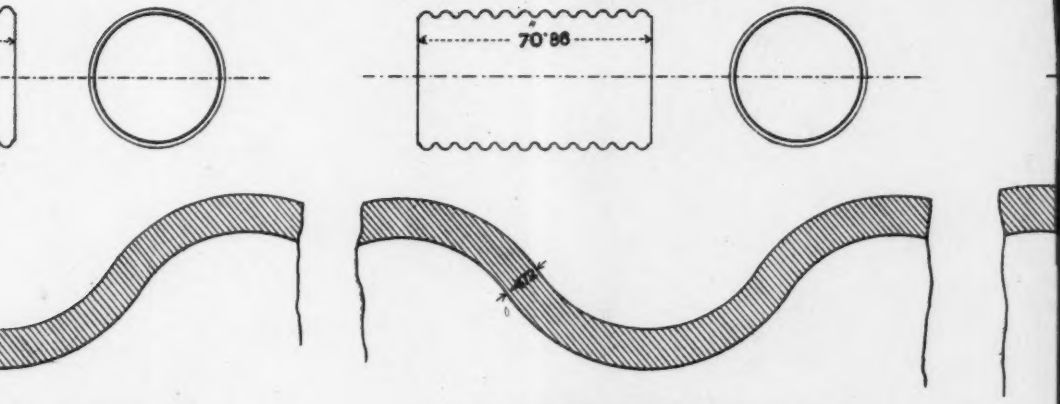


FIG. 16.



MENT No. II.	
DIAMETER - -	41·37
ETER - - -	37·20
OF STEEL - -	29·4
EL - - - -	28½%

EXPERIMENT No. III.	
GREATEST OUTSIDE DIAMETER - -	41·29
LEAST INSIDE DIAMETER - - -	37·24
TENSILE STRENGTH OF STEEL - -	21·1
ELONGATION OF STEEL - - - -	30%

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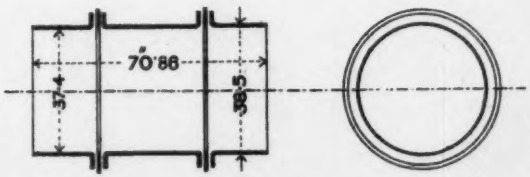


EXPERIMENT No. VI.	
GREATEST OUTSIDE DIAMETER - -	38·5
LEAST INSIDE DIAMETER - - -	37·4
TENSILE STRENGTH OF STEEL - -	22·8
ELONGATION OF STEEL - - - -	29·5%

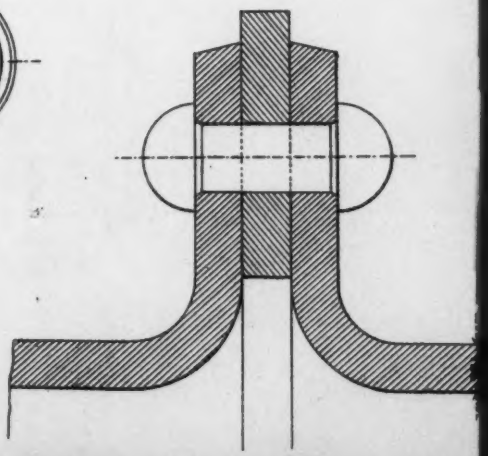
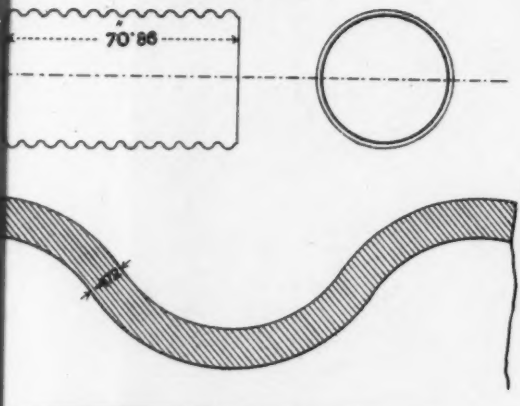
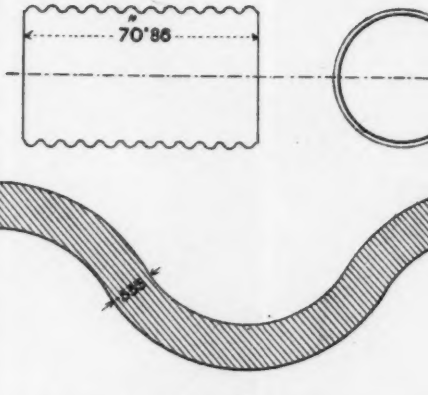


FIG. 16.

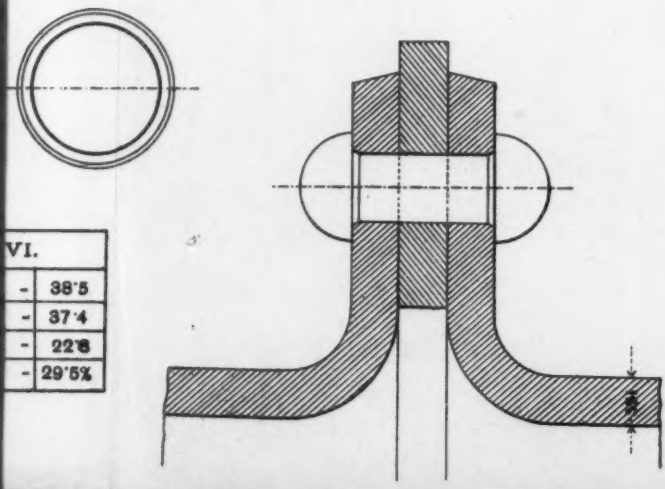


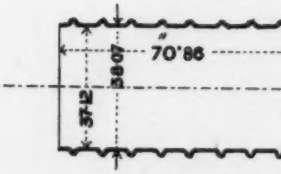


EXPERIMENT No. III.	
GREATEST OUTSIDE DIAMETER - -	41·29
LEAST INSIDE DIAMETER - - -	37·24
TENSILE STRENGTH OF STEEL - -	21·1
ELONGATION OF STEEL - - - -	30%

EXPERIMENT No. IV.	
GREATEST OUTSIDE DIAMETER - -	41·4
LEAST INSIDE DIAMETER - - -	37·6
TENSILE STRENGTH OF STEEL - -	22·0
ELONGATION OF STEEL - - - -	29%

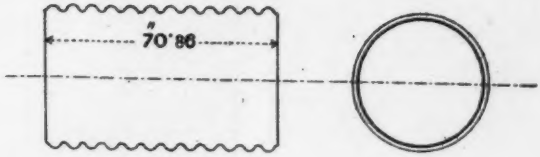
SHULZ KNAUDT, OF ESSEN, ON "THE LONGITUDINAL





VI.	
-	38·5
-	37·4
-	22·8
-	29·5%

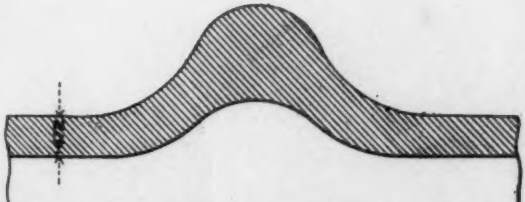
EXPERIMENT	
GREATEST OUTSIDE DIAM	
LEAST INSIDE DIAMETER	
TENSILE STRENGTH OF S	
ELONGATION OF STEEL -	



	41·45
	37·08
	22·05
	29½

EXPERIMENT No. V.	
Greatest Outside Diameter - -	41·45
Least Inside Diameter - - -	39·09
Tensile Strength of Steel - -	24·3
Elongation of Steel - - - -	26%

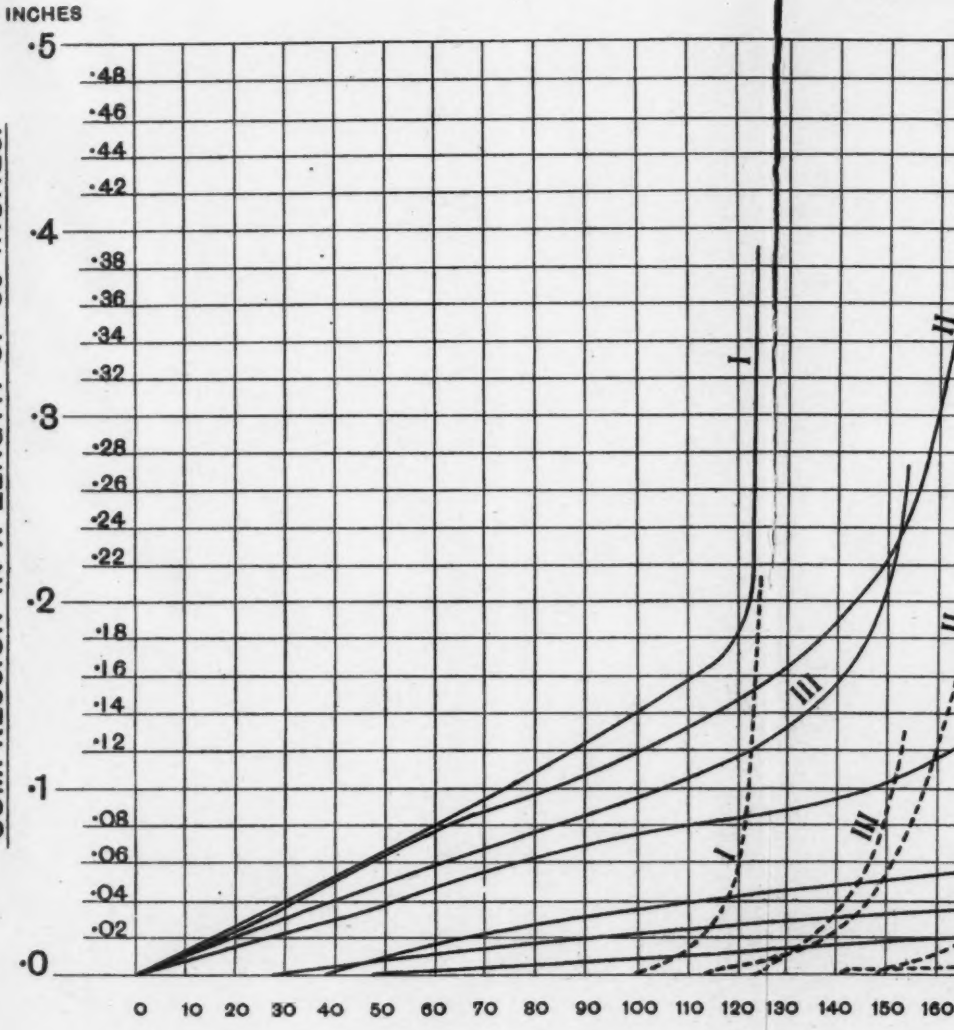
AL ELASTICITY OF FURNACES."



IENT No. VII.	
Diameter - -	38·07
eter - - -	37·12
of Steel - -	27·1
el - - - -	26%

EXPERIMENTS MADE B

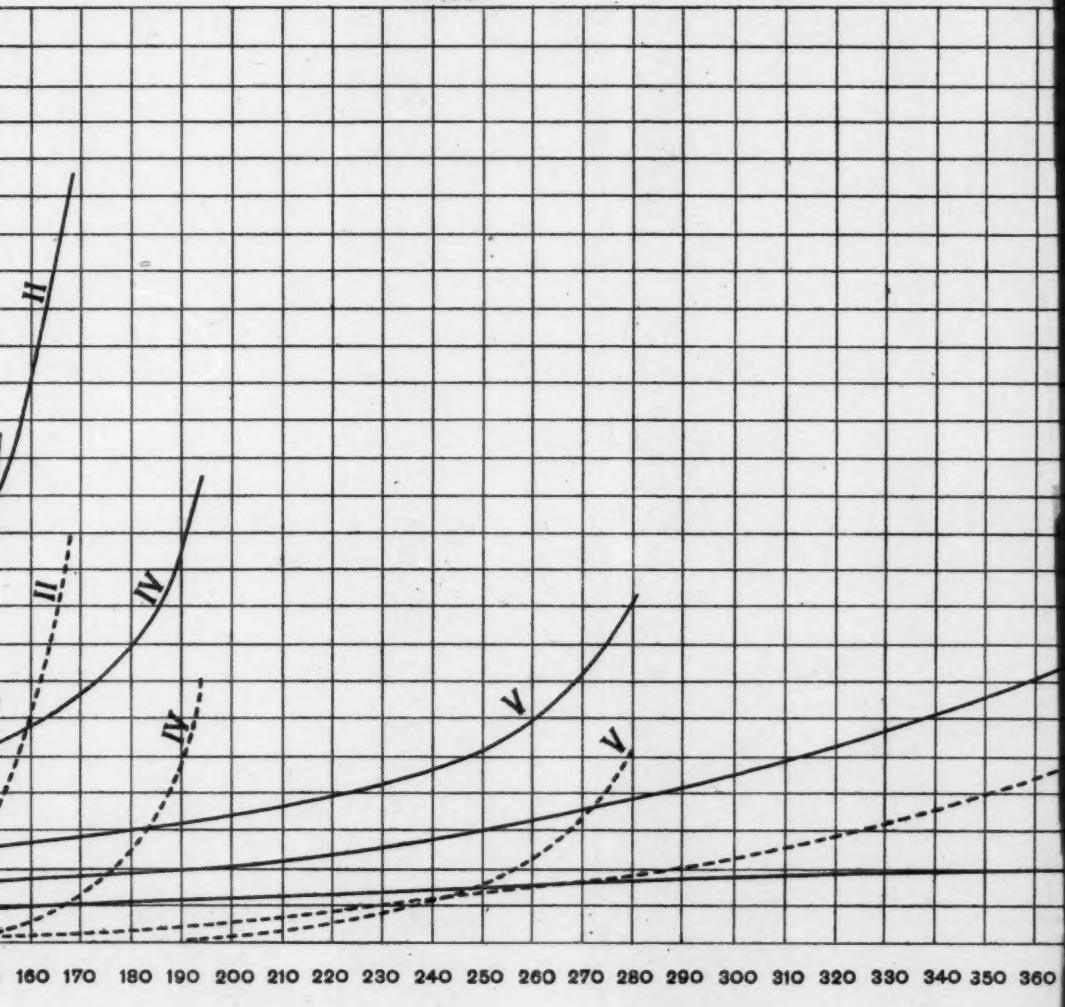
ON "



BY HERR VON KNAUDT, OF MESSRS. SHULZ KNAU

"THE LONGITUDINAL ELASTICITY OF FURNACES."

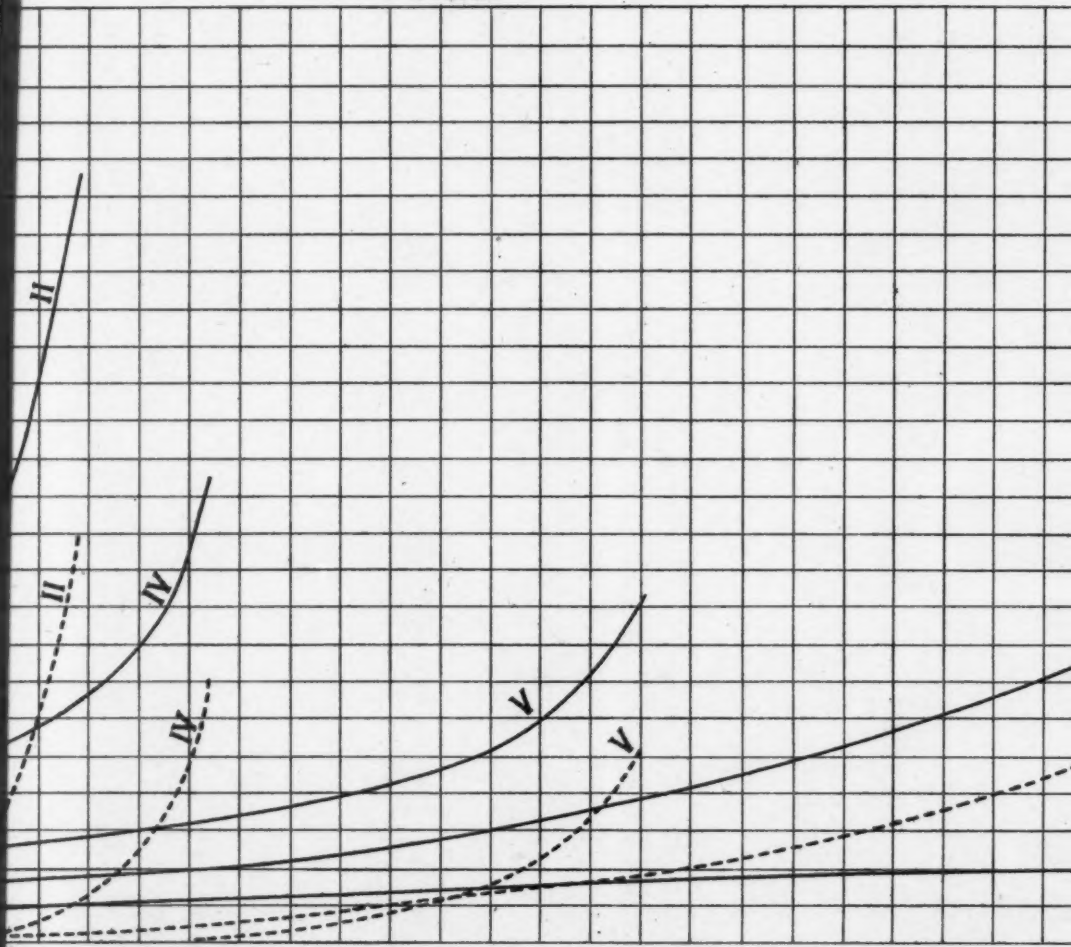
FIG. 17.



BY HERR VON KNAUDT, OF MESSRS. SHULZ KNAUD

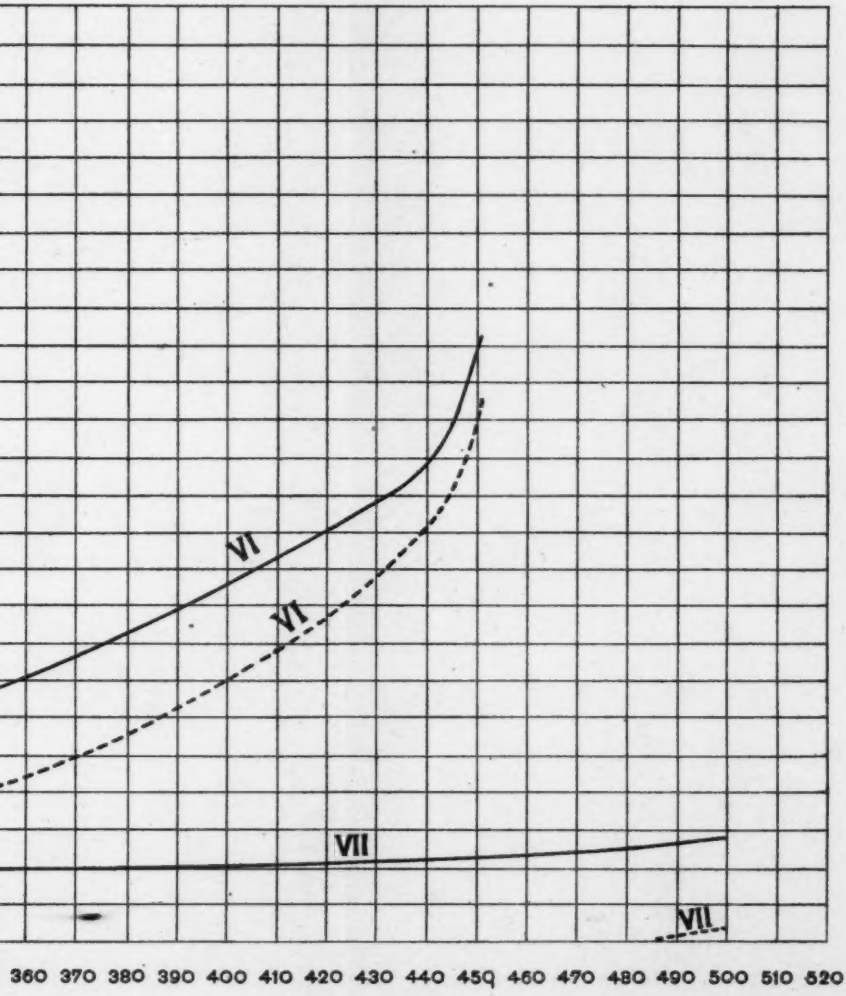
"THE LONGITUDINAL ELASTICITY OF FURNACES."

FIG. 17.



TOTAL PRESSURE ON END OF FLUE IN TONS.

AUDT. OF ESSEN,



Accidents to furnaces generally result from collapse; the development of small cracks along the line of fire; circumferential ripping in the body of the furnace; and the ripping at the back ends, usually starting from the flanged corners. Neglecting shortness of water, collapse of furnaces is generally due to accumulation, either of scale or of oily deposit, and may be prevented by the use of evaporators and feed-water filters. The development of small cracks along the line of fire is caused by local overheating of the materials due to undue accumulation of deposit. The inward projections of the Fox Furnace, for example, are towards the fire, and by receiving the greatest heat cause an undue accumulation of scale in the narrow pocket or cavity formed by the inward corrugation, with the result that the crowns of the corrugations get overheated and sometimes crack. These cracks also occur in the Purves, but generally on or near to the thickened ribs. The most serious form of cracking, however, is the sudden and unexpected ripping of a furnace, either circumferentially in the body or at the back end, these cracks sometimes extending for 18 inches, and completely destroying the furnace. This form of accident can, in the writer's opinion, be avoided by the design and manufacture of the furnace, the disposition of the stays and the suitability of material. It should never be lost sight of when designing the flanged end of a furnace that the more work put upon the material during its manufacture, the greater is the danger of ultimate failure by cracking. It is also known that certain movements of the furnace result from expansion and contraction, consequently if these movements are concentrated at points and not distributed uniformly over the length of the furnace, then, if the furnace cracks, it will be at the weakest point of concentration. Accidents of this nature have also been caused by the leakage of feed-circulating devices fixed in the boiler bottom. The disposition of stays also appears of importance, as a furnace must and will move; consequently, if the tubes are kept too low, the saddle short, or the furnace stayed in all directions, with stays arranged very near to the surface of the furnace, the very feature of the utmost value, viz., longitudinal elasticity, is

entirely neutralized. If it is desired to surround a furnace with stays they should, in the writer's opinion, be a slack fit longitudinally. Regarding material, it has been seen that high tensile steel does not remain ductile so long as low tensile steel; also, that the tensile strength when under working conditions is greater than when cold; further, at the temperature of working condition the steel is at the point of uncertainty, or the point of the greatest brittleness, consequently the writer would again say that steel not exceeding 26 tons tensile strength is more suitable for furnaces than steel not exceeding 30 tons.

It is also certain that the highest quality of steel of any tensile strength cannot be obtained without the use of the best materials, and excess of phosphorus and sulphur should never be present in steel which has to be subjected to such severe treatment as in a furnace. The design of flanged corners of the saddle and also the radius between the saddle and the furnace would also seem to effect the ultimate result. If uniform elasticity of design is essential to ultimate success, then these curves should be easy and not sharp and abrupt. These are all practical questions, however, upon which a broad discussion will be of the utmost value.

In conclusion, the writer desires to thank the various firms who have placed data at his disposal, viz., Messrs. T. Richardson & Sons, of Hartlepool; the Leeds Forge Company, of Leeds; Sir John Brown & Co., of Sheffield; the Farnley Iron Company, of Leeds; Messrs. Howden & Co., of Glasgow; Messrs. J. & H. Holmes & Co., of Hull; and Messrs. Hall, Russell & Co., of Aberdeen. The paper by Herr Von. Knaudt, of Essen, has supplied the valuable experiment in the longitudinal elasticity, and the temperature tests by the United States Government are certainly the most accurate and most valuable ever compiled, and reflect the greatest possible credit on the officials by whom they were made.

XU

TABLE I.—FURNACES MADE WITH ADAMSON JOINTS.—Tests made at the Works of Messrs. Hall, Russell & Co., Aberdeen, in 1882, and Messrs. J. Howden & Co., Glasgow, in 1887.

Date of test.	Length of furnace.	No. of rings.	Thickness of plate in inches. Front end.	Middle.	Back end.	Mean.	External diameter in inches over plain part.	Greatest difference in diameter at any part.	Collapsing pressure.	Collapsing coefficient $\frac{P \times D}{T}$.	Ultimate tensile strength of steel.	Collapsing coefficients reduced to steel of 27 tons tensile.	Position of collapse.
1882	6 feet 5¾ inches total; length of rings, 18¼, 19, 19 and 20 inches.	4	...	* ...	...	1st ring, ⅜″ 2d ring, 11/32″ F. 3d ring, 5/16″ 4th ring, ⅜″ B. .546	43	2/64	3d ring collapsed at 700.	64,213	*	61,918	...
1887	7 feet 0½ inches total; length of each ring, 23 inches.	4	...	...	...		43.09	No record.	1st ring, 840. 2d " 760. 3d " 840. 4th " 835. Mean, 814.	64,240	*	61,945	...

The collapsing coefficients are calculated on the external diameter of the furnaces over plain part.
* No record; probably 26 to 30 tons; assumed to be 28 tons.

TABLE II.—FOX'S PATENT CORRUGATED FURNACE.—Official Tests made at the Leeds Forge, Leeds, in 1882, 1890 and 1891.

Date of test.	Length of furnace.	No. of corrugations.	Greatest length of flat at end.	Thickness of plate in inches. Front end.	Middle.	Back end.	Mean.	Mean diameter in inches.	Greatest difference in diameter at any part.	Collapsing pressure.	Collapsing coefficient $\frac{P \times D}{T}$.	Ultimate tensile strength of steel.	Collapsing coefficients reduced to steel of 27 tons tensile.	Position of collapse.
	Ft. In.													
November, 1882.	6 9	13	...	...	...	...	.52	35.875	...	900	62,091	22.7	73,852	
November 4, 1890, and Feb. 11, 1891.	6 3 15/16	11	5⅞	.339	.364	.331	.349	34.125	5/32	830	81,157	29.05	75,430	On flat.
" " ..	6 3¾	11	6	.345	.384	.398	.378	34.25	5/32	800	72,486	29.17	67,094	* On flat.
" " ..	6 7	12	6	.419	.463	.462	.452	33.625	5/32	1,130	84,062	29.26	77,569	Second corrugation.
" " ..	6 8 1/16	12	6 5/16	.471	.473	.454	.468	34.469	5/32	1,090	80,280	29.02	74,692	* On flat.
" " ..	6 5½	11	6 1/16	.577	.585	.551	.574	33.593	3/32	1,400	81,934	28.55	77,485	11th corrugation.
" " ..	6 6 1/16	11	6¼	.542	.582	.596	.575	34.937	3/32	1,410	85,671	29.41	78,650	* Front corrugation.

TABLE III.—FARNLEY SPIRALLY CORRUGATED FURNACES.—Official Tests made at the Works of the Farnley Iron Co., Limited, Farnley, near Leeds.

May, 1888.....	6 5⅜	...	...	...	...	...	.559	39.231	...	835	58,601	‡28	56,508	No record.
"	6 6⅛	...	...	...	...	...	.548	39.132	...	850	60,697	‡28	58,529	"
"	6 6⅛	...	...	...	...	...	.442	38.928	...	670	59,008	‡28	56,900	"
"	†6 6⅛	...	...	...	...	...	.446	39.394	...	570	50,346	‡28	48,548	"
"	6 6 3/16	...	...	...	...	...	.354	39.296	...	515	57,168	‡28	55,126	"

TABLE VII.—MORISON'S SUSPENSION FURNACE.—Official Tests made at the Leeds Forge, Leeds, in 1891.

Sept. 25 and 26, 1891..	6 5½	9	4⅜	.340	.310	.340	.325	34.156	11/16	795	83,550	27.64	81,615	Flat at front end
" " ..	6 7¼	9	4½	.341	.391	.368	.373	34.265	11/16	900	82,677	27.34	81,648	2d corrugation from front.
" " ..	6 5¼	9	4⅜	.445	.492	.451	.470	34.444	11/16	1,100	80,613	26.92	80,852	Uniformly on weld.

Sept. 25 and 26, 1891..	6 5½	9	4½	.340	.310	.340	.325	34.156	1½	795	83,550	27.64	81,615	Flat at front end
" " ..	6 7½	9	4½	.341	.391	.368	.373	34.265	3/32	900	82,677	27.34	81,648	2d corrugation from front.
" " ..	6 5½	9	4½	.445	.492	.451	.470	34.444	1 1/16	1,100	80,613	26.92	80,852	Uniformly on weld.
" " ..	6 6½	9	4½	.452	.458	.441	.452	34.687	1/32	1,050	80,578	26 56	81,913	7th, 8th and 9th corrugation at weld.
" " ..	6 7½	9	4½	.535	.516	554	.530	34 719	3/32	1,300	85,159	27.52	83,550	1st, 2d and 3d corrugation.
" " ..	6 7	9	5	.556	.573	.565	.567	34 078	1/32	1,340	80,537	27.09	80,269	Flat at front and 1st corrugation.

The collapsing coefficients are calculated on the mean diameter of the furnaces.
§ All these furnaces were annealed in the presence of the Board of Trade officer
* These three furnaces were annealed. † This furnace was not so true as the others. ‡ Assumed.

TABLE IV.—HOLMES' CORRUGATED FURNACES.—Official Tests made at Messrs. Charles D. Holmes & Co., Hull, in 1891.

Date of test.	Length of furnace.	No. of corrugations.	Greatest length of flat at end.	Mean thickness of plate.		Diameter outside of plain part.	Greatest difference in diameter at any part.	Collapsing pressure.	Collapsing coefficient $\frac{P \times D}{T}$.	Ultimate tensile strength of steel.	Collapsing coefficients reduced to steel of 27 tons tensile.	Position of collapse.
				Plain part.	In corrugation							
	Ft. In.											
1891.....	7 0	4	No record.	.515	.458	35.37	1/32	950	65,245	27.3	64,528	One corrugation and two adjacent plain parts.
"	7 0	4	"	.452	.424	35 62	1/32	750	59,104	27.5	58,029	Plain part between two corrugations.
"	7 0	4	"	.557	...	35 5	1/32	920	58,635	26.8	59,072	Plain part and two adjacent corrugations.

The collapsing coefficients are calculated on the diameters of the furnaces over flats.

TABLE V.—PURVES' PATENT FURNACES.—Official Tests made at the Works of Sir John Brown & Co., Sheffield, in 1886 and 1887.

Date of test.	Length of furnace.	No. of corrugations.	Greatest length of flat at end.	Thickness of plate in inches.				Diameter over flats in inches.	Greatest difference in diameter at any part.	Collapsing pressure.	Collapsing coefficient $\frac{P \times D}{T}$	Ultimate tensile strength of steel.	Collapsing coefficients reduced to steel of 27 tons tensile.	Position of collapse.
				Front end.	Middle.	Back end.	Mean.							
	Ft. In.													
March 12, 1886.....	6 6⅜	7	13⅛	.561	.582	.548	.568	38.61	1/32	740	50,302	27.2	49,932	On flat.
" "	6 6⅜	7	13⅛	.544	.554	.532	.546	38.685	1/32	760	53,847	26.6	54,656	"
1887.....	6 7⅛	8	7⅛	.299	.286	.310	.295	38 652	3/32	650	85,165	25.9	88,782	Between first and second ribs.
"	6 7⅛	8	7⅛	.327	.302	.313	.311	38 591	3/32	635	78,795	27.9	76,253	Between sixth and seventh ribs.
"	6 7⅛	8	7⅛	.482	.535	.483	.509	38.83	1/32	800	61,029	26 1	63,133	From end to end.
"	6 7⅛	8	7⅛	.529	.589	.529	.559	38 868	1/32	800	55,625	27.9	53,831	" "
"	6 7⅛	8	7⅛	.583	.632	.597	.611	38.847	1/32	875	55,632	27.6	54,422	" "
"	6 7⅛	8	7⅛	.576	.638	.600	.613	38.788	1/32	873	55,239	28.3	52,701	" "

TABLE VI.—PURVES' PATENT FURNACES.—Official Tests made at Sir John Brown & Co.'s Works, at Sheffield, in 1889.

1889.....	No record	No rec	9½	No record.	No record.	No record.	.307	38.78	No record.	675	85,265	28.8	79,935	No record.
"	...	...	9⅜	...	...	...	.362	38.70	...	700	74,834	28.0	72,161	"
"	...	...	9⅜	...	...	...	.461	38.70	...	870	72,034	27.3	72,231	"
"	...	...	9⅜	...	...	...	.466	38.72	...	950	76,935	29.1	72,738	"
"	...	...	9½	...	...	...	.585	38.63	...	1,065	70,326	28.7	66,160	"
"	...	...	9⅛	...	...	...	.578	38.65	...	1,145	76,564	27.4	75,446	"
December 23, 1890...	...	...	...	...	...	....	.522	38.75	....	1,020	75,718	....	....	"

The collapsing coefficients are calculated on the diameters of the furnaces over flats. Corrugations spaced 9 inches apart.

STEEL CASTINGS.

By George D. Strickland, Passed Assistant Engineer, U. S. Navy.

After the able article of P. A. Engineer Ira N. Hollis, U. S. N., on this subject, published in the Journal for August, 1889, there remains but little that may be offered of any real interest or value. However, it may be stated that there has been a marked advance in the manufacture of steel castings; the greater part of them may be depended on as being sound; the surfaces are smoother and cleaner; a uniformly higher grade of metal is made; the cost to the manufacturers and to the users is much less, though the decline in the prices for castings has been more rapid than the reduction in the cost of manufacture; and certain changes in or modification of the methods then employed have been found necessary.

The increased demand for steel castings has been met by the several new Steel Casting Plants which have been put in operation during the year 1892. That all have been well satisfied with the amount of work they have had in hand, and that it has been necessary to enlarge one or more of the older plants, shows most conclusively a growing disposition to use steel castings. Nevertheless, it is true that engineers still lack the confidence which castings of steel by the open hearth process certainly merit. In view of this lurking suspicion or distrust (and in many instances prejudice) it is but fair to the manufacturers to state, and should be of interest to the users to know, that, during the last year, great improvements have been adopted in the methods of making, and in the quality and appearance of the castings. These improvements have been brought about by the exactions of the Government requirements and of the trade; by competition; and by the consciousness of the manufacturers

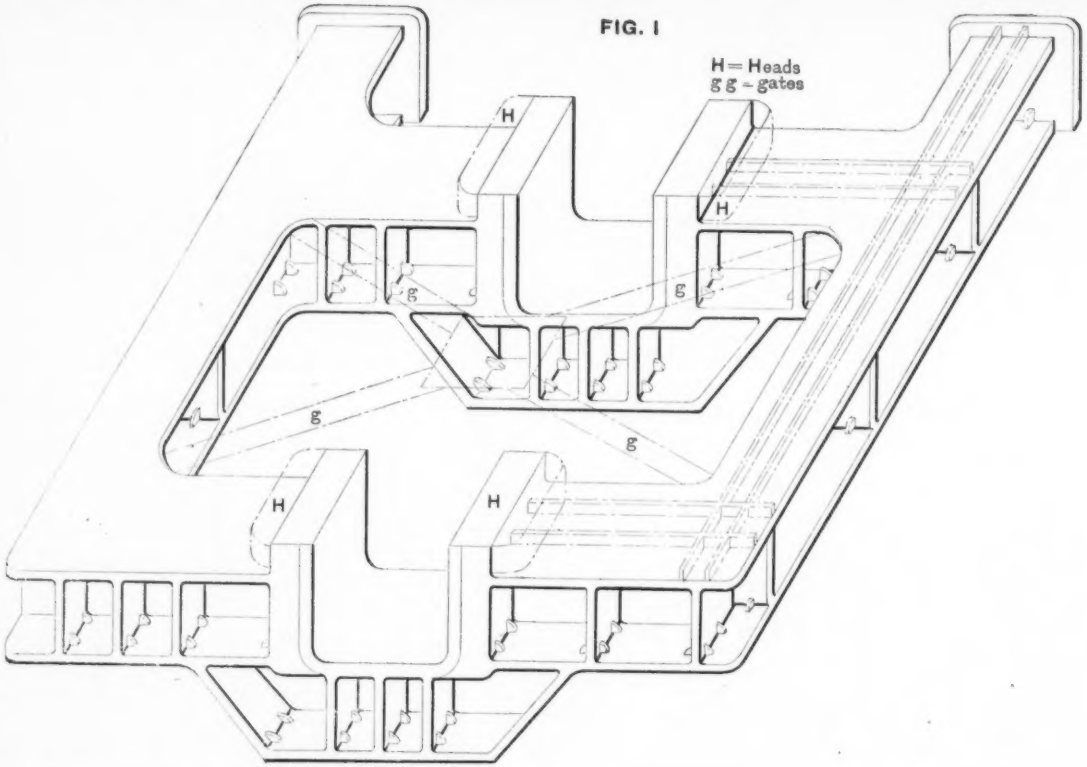
that defects can be traced to their causes and remedied, though only by close study and observation, careful and intelligent workmanship, the use of purer stock and by experiment. While it has undoubtedly been to their individual interests to do so, it has nevertheless been gratifying to observe the efforts in these particulars by the manufacturers with whom the writer has been associated.

It is an interesting fact that thousands of tons of steel castings have been made during the past year, of which an exceedingly small percentage, even as compared with the previous year, has been rejected owing to surface defects or imperfections developed during machining. There has also been a greatly reduced percentage of loss in the foundry, which now does not reach one-fourth of the amount one year ago. This great weight of castings shipped during the year was made up of all the varied shapes which entered into the building of hulls and machinery of the cruisers and battleships for the Navy, and into a diversity of other more or less important structures for the trade (including blowing-engine crank-shafts 17 feet long and 25 inches diameter, weighing 40,000 pounds), with weights ranging from 50,000 pounds to 1 pound. In view of all these facts, it is certainly not going too far to say that all concerned should feel greatly encouraged, and that, when correctly designed, steel castings can be successfully made, and may be much more generally used without any feeling of doubt as to the strength of the machine into which they may be incorporated.

There is no hesitancy in accepting proposals for castings of intricate design, which may now be made without the expensive precautions believed a few months ago to be absolutely necessary, when, at the same time, it was reasonably expected there would be a large number of failures. Many of these shapes would not then have been entertained by the manufacturers.

The physical results are uniformly higher, ranging from 62,000 to 70,000 pounds tensile strength, with an elongation of 15 per cent. to 25 per cent. in 8 inches, and 1-inch by 1-inch square bars bending through 90° to 180° without fracture over a radius of one and one-half times the thickness of bar. The Gov-

FIG. I



ernment requirements of sixty thousand pounds minimum tensile strength per square inch, an elongation in 8 inches of 10 per cent. to 15 per cent., and 1-inch by 1-inch square bars bending through 90° and 120° without fracture, over above-mentioned radius (castings and specimens annealed), are obtained without difficulty. There is also maintained a greater uniformity in the physical qualities of the heats, and it is seldom indeed that the tensile strength drops to the Government's minimum above given, or that there is any deficiency in elongation or bending when annealed.

DESIGN.

Many failures may be traced directly to the designing room, unless the manufacturer has taken the precaution to examine carefully the patterns before undertaking the execution of the order. To neglect this, when so little is generally known of the necessary changes from designs for iron castings, is to insure for themselves losses which otherwise would have been avoided.

In steel castings there should be as great uniformity of thickness as possible, which will greatly assist in the elimination of shrinkage cracks and cavities. To illustrate designs which meet these requirements atttention is called to Figs. 1 and 2, representing bed-plates and engine housings for *Protected Cruisers Nos. 12* and *13*, which are excellent examples. The loss of any castings from these patterns was due to carelessness of workmen.

It is necessary to avoid sharp interior angles, introducing instead fillets of ample radii. Mr. Hollis drew attention to the importance of this, and we have had numerous opportunities to verify the correctness of his statement. After repeated failures, the interior angles in pistons for *Cruisers Nos. 7, 8, 9, 10* and *11* were filletted, and the defects at inner edges of follower seats eliminated. Usually, in cases of this kind, the imperfections are not detected until during machining, being beneath a thin layer of sound metal; this layer is often so thin that it is removed when the castings have been annealed at a scaling temperature, when, after cleaning, many are found porous and worthless, which, before annealing, presented perfect surfaces.

The patterns furnished of pistons for the *Protected Cruisers Nos. 12* and *13* (Fig. 3) and *Battleships Nos. 1* and *2* met the last-mentioned requirement perfectly. There were sixteen pieces in all, of diameters from 42 to 92 inches. None were lost in the foundry or rejected on surface inspection, nor did any defects develop during machining. Three of 92 inches diameter would not finish to size, and were rejected at the builders' works, after which $\frac{5}{8}$-inch was added to face of pattern at *b*. The pistons were to finish to 3 inches at the point shown, but on boring through the pattern, it was found to measure but $3\frac{1}{16}$ inches. These rejections, therefore, were due to a mistake made by the pattern maker. They were all cast as shown, the method of heading being similar to that shown by Mr. Hollis in his Fig. 1, except that there was but one head used, and of a cross section equal to hub of piston, being really an extension of the same. The manner of admitting the metal gave it an opportunity to wash the mold, and drive out the air and gases in its progress.

For pistons of this form (Fig. 3) heading and gating, as shown by Mr. Hollis, was abandoned, it being found that the full influence of heads did not reach the masses of metal at the hubs, and the chances of imperfections in the flanges were increased. Comparing Fig. 3 with Fig. 4, the natural inference is that the present method used is the correct one.

The cause of failure of Fig. 1 (Mr. Hollis' article) was probably due to the method of introducing the metal, the gates being underneath instead of on edge of flange.

PATTERNS.

The designs of the castings being satisfactory, it is necessary in making the patterns to "part" them with a view to location of heads and gates rather than to the convenience of molding, and it would not be unwise for purchasers, when possible, to obtain and use the suggestions of the successful bidder as to the best manner of making the divisions and the necessary amount to allow for shrinkage, which might avoid extensive alterations or modifications involving considerable delay and expense.

A good, practical manufacturer will hesitate to attempt cast-

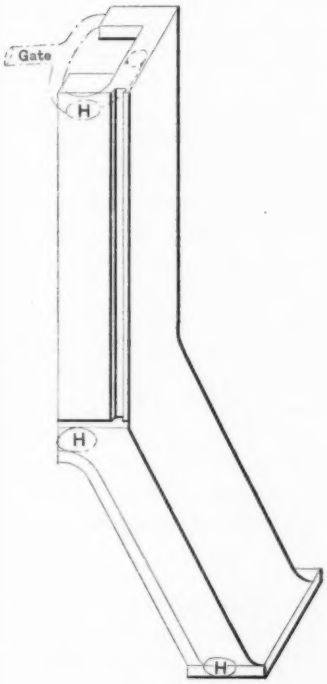


FIG. 2

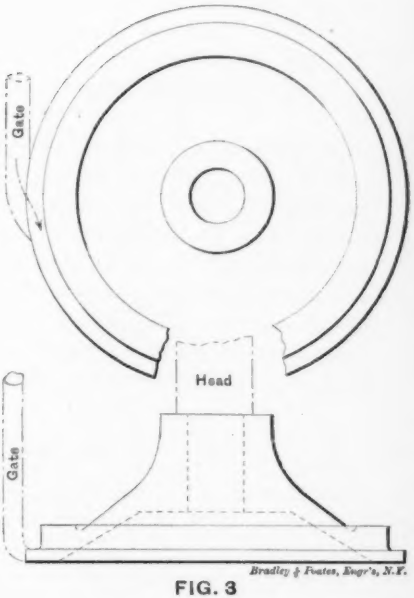


FIG. 3

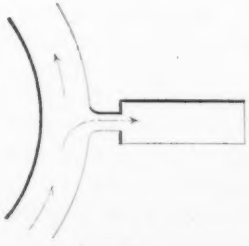


FIG. 9

FIG. 10

FIG. 11

ings from patterns furnished if his experience has taught him that changes are advisable before sound castings can be assured, while there are others who will undertake anything, and trust a good deal to luck; but, by such trials, without the essential alterations, considerable loss is incurred by the manufacturer and no little annoyance to the user from delay in delivery.

Probably the best guide at hand for allowances for shrinkage is the following card, issued by the Standard Steel Casting Co., of Thurlow, Pa.:

"Shrinkage Varies According to Shape of Casting.—Allow about three-sixteenths ($\frac{3}{16}$) inch per foot. For round, solid bodies, about one-quarter ($\frac{1}{4}$) inch per foot. Cores or inside openings generally do not shrink; therefore, no allowance is made. Core boxes, one-sixteenth ($\frac{1}{16}$) inch to three-sixteenths ($\frac{3}{16}$) inch, according to size, smaller than core prints. Place large fillets in all corners. Castings to finish, allow one-quarter ($\frac{1}{4}$) inch; this to be increased for large castings. Mark pattern where castings are to be finished. Patterns should be well supplied with rapping plates."

MOLDS AND CORES.

To be thoroughly competent to prepare molds for steel castings, it is not only necessary to be a good molder but a mechanic. The molds and cores, when in condition to receive the metal, have surfaces almost as hard as porcelain, though easily broken by a light blow or a moderate jar in handling. Owing to the nature of the surfaces, the several parts must fit together much the same as in the assembling of a machine, every piece having been finished to size and admitting of but slight alterations with the file. As one man makes the molds and, usually, another the cores, and possibly a third "builds up" on the floor preparatory to pouring, the molds are placed at some disadvantage as compared with the machine; not so much because of the several different workmen interested in the finished molds, as owing to the complicated nature of some, and to the fact that the different pieces may swell or shrink unequally, and when finally assembled and the pieces cast the resulting casting may

not be true to pattern. Although this does not frequently occur, it has happened, and invariably in expensive shapes. Should it be found while "building up" that the molds and cores do not conform to dimensions, it is not even then, as in loam or green sand, possible to prepare the molds for that heat, as the surfaces will not admit of cutting and building up without after-drying.

A molder should be able to determine whether or not good castings can be made from the patterns presented without first making trials. With some radically new shapes it may not be possible to determine this, but a thoroughly competent person would be able to make a close approximation to the actual results likely to be obtained.

There is no guess work admissible in a well-conducted steel foundry; everything must be executed on sound principles, or the percentage of losses will be surprisingly high.

The manner of making molds and cores, and the materials which enter into them, is of the first importance, as they greatly influence the soundness of the castings and the condition of the surfaces.

The faces should be able to resist the intense heat of the molten metal and scoring effect during pouring, thus avoiding to a very considerable extent rough and scabby surfaces and sandy castings.

It is important that facing sand should be high in silica and free from iron. The presence of iron causes the sand to fuse to the surfaces of the castings, resulting in roughness. This effect may, to a great extent, be eliminated by the use of a good wash.

The molds and cores should be sufficiently strong to support the ferrostatic head necessary to insure soundness, but frail enough to crush during contraction, while of a sufficiently open nature to permit of a ready escape of the air and gases. This latter chiefly applies to the air which was contained in the molds and the gases evolved by the contact of the hot metal with the mold surfaces, as the gases evolved from the metal before or while solidifying have but little opportunity of exit through this medium, for the reason that the metal almost at once chills on the surfaces of the molds and cores, as stated below.

All of these precautions, which have so much to do with the final conditions of the castings, will be of no value if the molds are not thoroughly dry.

The temperature during drying should be raised very gradually. The copes of molds are sometimes drawn out of shape by the shrinking of the wooden brackets, which are necessary to hold the sand, and which are burned out in the drying, as is the sawdust mixed with core material. By this, additional means is given the gases for escape.

In making molds the sand is tamped from what is, when finished, the bottom of drag and top of cope. This permits of the sand being packed solid under the patterns, giving firmness where most needed. When this operation is completed the bottom plate is securely fastened in place, the flask turned over and the pattern drawn.

In many long and difficult shapes, such as sternposts, heavy mill gearing, "wild cats," etc., the cores are built up in the molds and dried separately. It is necessary to do this that the patterns may be drawn. The cores are built up and removed, which leaves the patterns free.

As the metal sets almost immediately on entering the molds, and, irrespective of size and shape of casting, the importance of having the mold hot at time of pouring will be understood; avoiding in great measure a blowy surface, or rather that part of the casting just underneath the surface, which so frequently develops during machining, or often after annealing at a scaling temperature as hereinbefore mentioned. The nature of such defects Mr. Hollis alludes to in his "First Provision to Secure a Good Casting," p. 187, August JOURNAL, 1889. This is believed to apply particularly to castings of small cross section as compared with their surfaces, they being much more frequently found defective in this particular than heavy ones in the same heat and with molds made of the same material and in the same manner. These blow holes, which are usually quite small, are thought by some to be caused by an excessive casting temperature, from which we might infer that it would be of advantage to use a lower casting temperature for such shapes. There are

others, however, having an extended experience in the making of light castings who assert that their greatest success has accompanied high casting temperatures.

In evidence of the statement that the material chills on the faces of the molds almost immediately on coming in contact, may be offered the notes taken after the loss of a heavy casting by the breaking of a flask, which happened when the mold was about three parts filled. The drag not being firmly and uniformly seated, the bottom plate suddenly broke, and within a couple of seconds the contents of the mold were emptied on the foundry floor. On removing the cope one end of the casting appeared to be all right, but on examination was found to be but a shell of about one-sixteenth inch in thickness, and this thin layer or shell was also found to extend to the full height the metal had reached.

Advantage might possibly be taken of this rapid surface cooling of steel and turned to account in the manufacture of light ornamental castings.

HEADS AND GATES.

An intelligent arrangement of heads and gates is absolutely necessary. With the metal all that could be desired, the castings would probably be imperfect if this should be neglected, and the imperfections might or might not show on the surface. As almost every different shape requires a special treatment in this particular and a study and comparison with castings already made of more or less similar forms, the subject can only be properly treated by those having an extended experience.

It is by no means necessary that all castings should be headed to insure their soundness. On some shapes the heads would really be a detriment rather than a benefit, robbing the castings instead of feeding them, resulting in lighter castings than without the use of heads. This fact is certainly pointing in the right direction. Every step toward simplicity and reduced expense in moulding and machining ready for shipment is a benefit to be shared alike by manufacturers and users.

Again, some shapes are more solid when "lumping" is re-

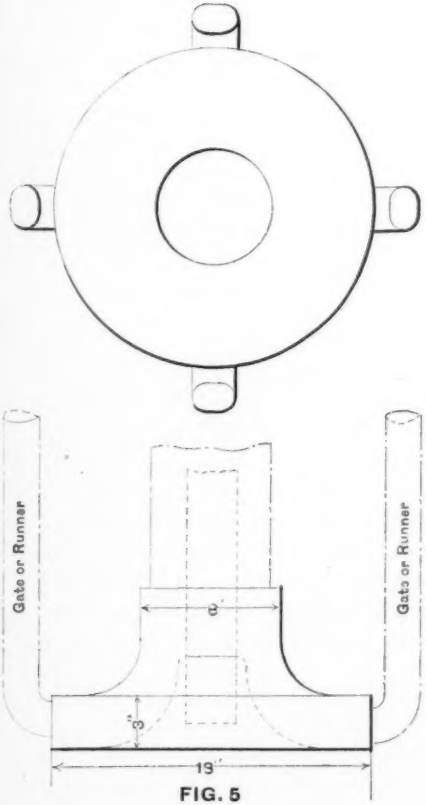


FIG. 5

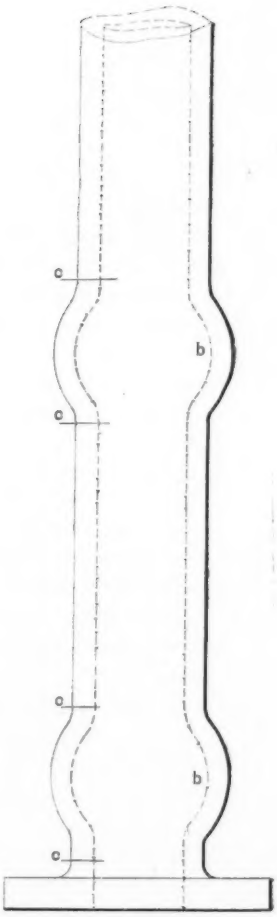


FIG. 12

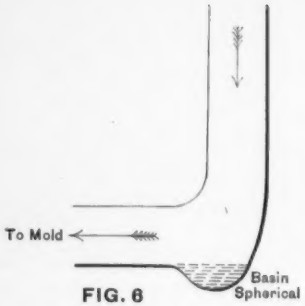


FIG. 6

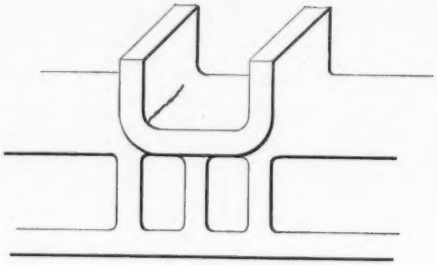


FIG. 7

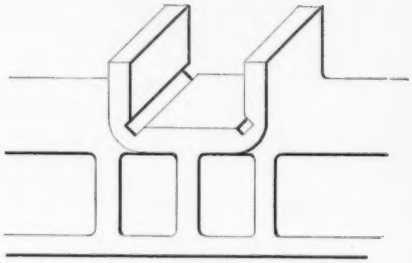


FIG. 8

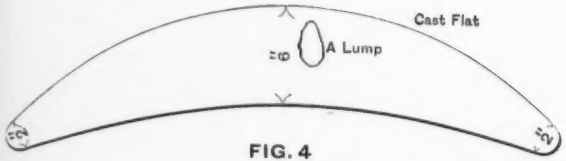


FIG. 4

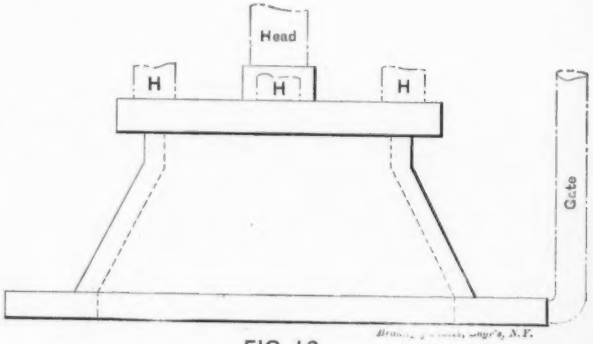


FIG. 13

sorted to, instead of heading, as shown in Fig. 5, full lines representing the "lumps" and the dotted lines the heads. With the head, its influence is not fully felt at the rim, while with four "lumps" the web is kept alive until the whole casting is satisfied. With "lumping" no gates, as such, are necessary, the metal being poured in one or more of the "lumps." The principal drawback to the method is that any loose material that may be in the mold is likely to be distributed throughout the casting, while with the head, if correctly gated, the metal, while pouring, has a tendency to wash the mold and drive the foreign substances to the head.

The practice of fixing plates of iron at the bottom of high gates—such as are necessary in the pouring of a long casting on end, and mentioned by Mr. Hollis—has been discontinued, and has been superseded by the thorough nailing of that part of the gate; but even nailing is now a thing of the past, and the present practice is to form a basin of facing sand with sloping sides which prevents the falling metal from striking a normal blow, and the basin retains sufficient metal to serve as a cushion. This method has stood the test of molten metal falling through twenty feet. With improved washes, nailing the surfaces of the molds is unnecessary.

While it is important to have the heads large enough to meet all the demands of the castings, it is not economy to have them unnecessarily so, for the reason that the heads become a stock of a lower grade, owing to their containing the impurities thrown off by the metal in the castings. The natural tendency is, therefore, to work near the minimum, and not infrequently heads have been reduced below this point and their efficiency lost.

The heads should be of ample height, that all piping may be above the castings.

BUILDING UP.

Great care is exercised in the assembling of the molds. They having been thoroughly dried, are kept as hot as possible and carefully examined that the surfaces are intact, solid and clean; carelessness in any one of these particulars being equivalent to

imperfect castings. When the molds are not perfectly dry the surfaces of the castings will probably be rough or scabby and blowy; if cold, imperfect in contour, and in small castings blowy; and if the walls of the molds are loose and surfaces dirty, the castings will be very likely to contain scabs and sand holes.

MELTING.

The selection of the stock and the manipulation of it in the furnace are naturally of great importance.

As all melters are not governed by the same circumstances in the selection of their stock, we may place them in three classes: Those who judge of the stock by its name, maker or trade grade, and the results obtained by them from certain mixtures of the same; those who are influenced by this in addition to their individual knowledge of the stock, gained by long experience, observation and experiments; and, finally, those who regulate their proportions chiefly by the chemical ingredients of the stock, and experiment.

The results obtained by the last mentioned are naturally superior to the productions of class 1. The only correct foundation being the chemical analyses, those governed by it must be in a better position to improve their metal than even the melters in class 2; at the same time, those of classes 2 and 3 usually obtain a metal of about the same quality, and can feel almost equally certain of the results of their respective heats. In proof of this, the writer's attention has been called to the experience of Mr. M. Mattieu Williams, given on page 188, in his work on the "Chemistry of Iron and Steel Making," where he says: "When engaged as chemist in the works of Sir John Brown & Co., of Sheffield, I made careful analyses of the numerous brands of pig iron that were used for the various purposes. The manager of the iron department was a remarkable example of those self-taught unconscious Baconian philosophers I have before alluded to. He had during many years been observing; experimenting and generalizing, his inductions consisting of a code of original rules for the manufacture of iron suitable for various purposes. In discovering the best 'blends' of pig iron for manu-

facturing, respectively, armor plates, rolls, boiler plates, angles, &c., * * among his other mixtures were certain 'blends' for making puddled steel. * * * He selected these, like all the others, without having, or pretending to have, any knowledge of their chemical composition. * * * Upon purely theoretical chemical grounds, I had determined that certain brands among those I had analyzed were the best fitted for making puddled steel, and was anxious to verify my theory. To have asked directly for the iron manager's secrets would have been unreasonable, and therefore I simply gave him a statement of the analyses of these particular brands, all arranged together, and called them 'steel irons,' adding that, for the best work, I supposed that he mixed with them a proportion of a certain foreign brand. 'Hush! don't talk so loud; I don't want these fellows to hear you. Who told you I used these?' was the substance of Mr. Jevons' reply. My theoretical and his practical selection proved to be exactly the same in result."

METALLURGY.

Manganese.—The Metallurgy of Steel, by Henry Marion Howe, 3d edition, vol. 1, p. 42, says:

"Manganese alloys with iron in all ratios, being reduced from its oxides by carbon at a white heat, and the more readily the more metallic iron is present to combine with it. It is easily removed from iron by oxidation, being oxidized even by silica, and partly in this way, partly in others, it restrains the oxidation of the iron, while sometimes restraining, sometimes permitting the oxidation of the other elements combined with it. It is also apparently removed from iron by volatilization. Its presence increases the power of carbon to combine with iron at very high temperatures (say 1400° C.), and restrains its separation as graphite at lower ones. By preventing ebullition during solidification and the formation of blow holes, by reducing or removing oxide and silicate of iron, by bodily removing sulphur from cast iron and probably from steel, by counteracting the effects of sulphur which remains, as well as of iron oxide, phosphorous, copper, silica and silicates, and perhaps in other ways,

it prevents hot shortness, both red and yellow. (It does not, however, counteract the cold shortness caused by phosphorus). These effects are so valuable that it is to-day well nigh indispensable, though admirable steel was made before its use was introduced by Josiah Marshall Heath.

"It is thought to increase tensile strength slightly, hardness proper, and fluidity, to raise the elastic limit, and, at least when present in considerable quantity, to diminish fusibility. It is generally thought to diminish ductility.

Silicon.—The solvent influence of silicon on hydrogen and nitrogen gases is utilized by its introduction in the form of ferro-silicon into the bath. A large percentage of these gases is therefore by it held in solution by the metal during solidification. It could be introduced in sufficient quantities to cause all of these gases to be held in this state, but at a sacrifice of tensile strength and ductility. The percentage used at present in steel castings has no detrimental influence, and by some it is thought that the amount could be considerably increased and with advantage.

Aluminum in small quantities is believed to have the same effect, as regards blow holes, as silicon and manganese, without carrying with it any of the evils of these substances. It makes the metal more fluid, and is therefore used more particularly in heats for light castings.

Phosphorus in steel in any appreciable amount is objectionable, though its evil influences are much less noticeable in low carbon than in high carbon steel. In steel low in carbon it very materially increases the tensile strength and elastic limit—more particularly the latter—and even does not, in many cases, interfere with the ductility. It is so unevenly distributed, however, that but little dependence can be placed in the metal containing it, particularly when subject to shocks, as in a large proportion of machinery and ordnance castings. Its total elimination from steel castings is desirable. In experiments on two test bars cut from the same coupon (phosphorus 0.1 per cent.), side by side, one gave excellent physical results, while its neighbor was utterly bad.

A series of tests made of open-hearth steel, cast, and containing .083 per cent. phosphorus, gave the following results: Tensile strength, 63,000 to 66,000 pounds per square inch; elastic limit, 36,000 to 37,000 pounds per square inch; elongation in 8 inches, 14.12 per cent. to 24 per cent.; and reduced area at fracture, 18.58 per cent. to 41 per cent.; bending, over radius $1\frac{1}{2}$ times thickness of bar, 40° to 135°. The phosphorus at fractures was from .075 to .105 per cent. The effect of shock was not ascertained.

Since there is no difficulty in procuring abundance of basic stock, of .025 per cent. in phosphorus, the present limit of this element might reasonably be lowered, and particularly for engine castings. This is all the more necessary from the fact that the laudable tendency of the manufacturers is to increase the carbon in their steels because of the greater liquidity of the metal when at pouring temperature, the seeming beneficial effect in reducing blow holes, and the greater range of physical qualities obtained by annealing.

Sulphur.—In connection with this subject attention is invited to "A New Process for the Purification of Iron and Steel from Sulphur," by E. H. Saniter, F. C. S. (JOURNAL for November, 1892, p. 757.)

In this process contact with lime gave irregular and imperfect results. A more readily decomposable body than lime was found necessary for rapid and satisfactory removal. The inventor finally "discovered that calcium chloride and lime, *i. e.*, the oxychloride, is a powerful desulphurizing reagent."

No. of experiment.	Time molten iron was in contact with the mixture.	Sulphur in iron.		Mixture used.
		Before treatment.	After treatment.	
	hours.	*per cent.*	*per cent.*	
4	$2\frac{1}{4}$	0.38	.03	Lime.
6	$1\frac{3}{4}$	0.30	.12	Lime.
15	$\frac{1}{2}$	0.52	Trace.	Lime, 90 %; Calcium chloride, 10 per cent.
16	$\frac{1}{2}$	0.42	"	

"The supply of iron being kept molten in plumbago crucibles in contact with the substance named," the foregoing comparative results are obtained.

The results show that the lime will in two hours remove much of the sulphur, and that with a mixture of calcium chloride and lime a half hour will completely eliminate it. It was on these last experiments that Mr. Saniter elaborated his process.

The treatment costs but about thirteen cents per ton, which figure may be reduced with an effective plant.

The inventor claims for his process, as applied to the basic open-hearth furnace, that this sulphur is removed along with the phosphorus, the value of which cannot but be appreciated.

POURING.

Pouring requires good judgment, as the form of a casting determines the rate. The molds for heavy and chunky shapes may be filled as rapidly as the gates will carry the metal, while with light and intricate forms the metal must enter the mold more slowly. When a great number of small castings are to be made at one pouring, it is necessary that the casting temperature be high, as a portion of this metal may have to remain in the ladle as long as forty or fifty minutes.

If the metal is not "dead" in the ladle at time of pouring, the castings will be blowy.

"SHAKING OUT."

The weight and proportions of a casting govern the time it is kept in the sand after pouring, while some light castings may be "shaken out" at once. With other forms, made up of heavy and light sections, it is necessary to relieve the lighter parts as early as practicable, but at the same time avoiding exposing any portion to the air; the object being to break up such sections of the molds and cores as prevent the castings from creeping during solidification. Should the molds and cores be weak enough to crush of themselves, this precaution would be unnecessary; but as it is not possible to construct them to such a nicety the necessity for thus relieving the castings is obvious.

Heavy solid pieces, such as rolls and dies, are kept in the flasks about twenty-four hours; the copes are then removed and the drags turned over. After leaving the castings thus free in the sand two days longer, they may be taken out without danger of cracking.

Some large castings are permitted to remain in the sand five or six days, while with small ones it depends on the time at the disposal of the foreman.

Too early exposure to the air causes internal stresses, which, if not destroying the value of the casting by the development of shrinkage checks or cracks (often so small as to be scarcely distinguishable), may be eliminated by annealing. Frequently, however, complete rupture takes place. Heavy and expensive castings have been lost by too hasty exposure, and more than one night watchman has been startled by sharp reports caused by the breaking of some such piece from shrinkage.

BLOW HOLES.

With careful workmanship and suitable material, castings may readily be made having clean, smooth surfaces and free from cracks, scabs and sand holes, and of excellent quality; but there remains for the manufacturers the problem how to avoid shrinkage and blow holes. Many heats are almost entirely free from these imperfections, so far as can be ascertained by machining and the breaking up of such castings as are imperfect in contour; in other heats a small percentage are blowy and contain shrinkage cavities under the head, while occasionally a really bad heat is encountered.

The causes of shrinkage cavities are pretty well understood, and with a good heat and adequate and intelligent heading, the evil may be greatly reduced. These holes or cavities may be found under the heads even when it is known that the heads are of ample capacity, and that they remained molten sufficiently long to have fully satisfied the casting. They may be caused by the metal solidifying irregularly, and in such a way as to shut off the supply to these individual spaces.

Castings of steel may be fed by the use of the rod, as in iron

castings, but in so doing air is liable to be pumped in and the escape of the gases interrupted, and while eradicating one evil another equally pernicious is encouraged.

In some castings of heavy sections, broken up for scrap on account of surface defects, internal shrinkage cavities have been found, and, as a rule, at or near the neutral axis, the cooling having, apparently, taken place uniformly from the outside; this would seem to be rather an advantage than otherwise, in lessening weight of castings without reducing their strength.

The uncertainty of the extent of some surface cracks or checks caused by shrinkage has usually led to the rejection of castings otherwise sound. The bed-plates rejected for *Cruisers Nos. 7, 8, 9, 10* and *11*, because of checks or cracks in flanges or webs, such as might have developed at *aa*, Fig. 1, were found when broken up to be sound in every other particular, and the cracks to have previously been explored to their full extent (two to five inches), while in other instances when the cracks were at heavy sections, and of small extent on the surfaces, such as might have appeared at *a*, Fig. 3, they developed into deep, ugly cracks and large shrinkage cavities. No defects of this nature having occurred in similar shapes during the past few months, shows an improvement in the method of manufacture.

In blowy castings these peculiarities have been observed: that in light sections, such as man-hole covers, the blow holes are tubulous and usually extend entirely through the castings; that in heavier sections, piston followers *Cruisers Nos. 7* and *8*, there were but few on the surface and quite small; immediately underneath they grew larger, irregular and more numerous, beyond these a zone containing a greatly reduced number, but of good size, while still deeper they increased in size and numbers; that in still heavier sections comparatively few on the surface, beyond which a zone of solid metal, followed in turn, at or near the neutral axis, by numerous large and irregular ones. These observations were made of pieces in different stages of machining and when broken up under the "drop."

Frequently castings are blowy on the mold faces and sound

next to the cores, for which condition it would seem the molds were responsible.

The more fluid the metal at time of pouring the more air it will probably carry with it into the mold, as the tendency of the stream is to break up and offer greater surface, but at the same time, owing to its liquidity, it should quickly liberate it. It is believed, therefore, that it is not the air and gases already in the mould, nor the air carried in while pouring, that is the fruitful source of blow holes, but the gases evolved during contraction, which make every effort to escape, but are caught and imprisoned, and, so long as the consistency of the metal will permit, one gas bubble joins another, thereby increasing the size of the blow holes.

ANNEALING.

The time of annealing and temperature varies considerably with manufacturers. The time of raising is 16 to 50 hours, maintained 6 hours to 6 days, and cooling 16 hours to 5 days, and with castings of the same weight and no material difference in the proportions of the chemical ingredients. The physical results cannot, therefore, be expected to be the same. It is not possible to state what the differences actually are, as specimens will differ in results, even when taken from the same casting without annealing, showing that the metal is not perfectly homogeneous.

The results given by Mr. Hollis show that the Government requirements of 15 per cent. elongation in 8 inches, a minimum tensile strength of sixty thousand pounds per square inch, and a one-inch square bar bending through 90 degrees without fracture over radius 1½ times thickness of bar, can easily be obtained from a good heat and without annealing, when the necessary precautions have been taken while the castings are in the sand; and as annealing is now one of the Government requirements for constructive castings, it is only necessary with this metal to raise them to a temperature sufficient to relieve any internal stresses that may exist and allow them to cool in a closed furnace.

With the increased demand of the Government, raising the

bending test to 120 degrees over the same radius, and with a higher quality of metal, more attention must be given to this most important treatment, and unfortunately it is the one thing which at present seems least understood by manufacturers, though the disposition is to avail themselves of the latest information on the subject. Its importance is at once given prominence by the facts that a slight difference of temperature at or about the point where the state of the carbon changes makes the greatest difference in the physical results; that the changes in strength, ductility and hardness are chiefly due to the corresponding changes in the condition of the carbon, and that these changes are said to take place at about a dark red heat (1292° Fahr. according to Pouillet). At this temperature the combined carbon has been found by investigators to begin to change its state, and to pass into what has been termed by Rinam "cement carbon." The temperature applied, therefore, in the treatment of castings depends upon the physical qualities desired, and ranges between the above-mentioned temperature and, say, 2000° Fahr., the latter being the approximate temperature at which the carbon has been found to pass almost entirely into the hardening state. The temperature depends principally on the amount of carbon contained, and that these changes are accompanied by the reduction or breaking up of the pre-existing crystals, which, by the way, will again appear and increase in size with the increase of temperature, and with a corresponding decrease in the physical properties. As slight hardening raises the tensile strength, excessive hardening will sometimes lower it; so in annealing, the tensile strength is generally reduced, but may be increased. It is, therefore, clear that there must be a critical temperature for every heat poured, beyond which the beneficial results of annealing are lost.

In annealing, the castings should be uniformly heated in a closed furnace.

The more slowly the castings are cooled to low visible redness, the more the toughness of the metal is increased, but below this temperature it has been found that the rate of cooling has but little effect upon the physical properties.

Small castings are frequently annealed by burying them in lime, ashes, or some other slow conductor. Lime appears to give the best results, and a small casting, when so treated, will require days to cool.

It has been stated that steel high in phosphorus is considerably increased in ductility and tensile strength by rapid cooling from the critical temperature. The results obtained from a limited number of experiments seem to agree with this statement.

The construction of an annealing furnace should be such as to give and maintain a uniform temperature throughout its necessary capacity, and to protect the castings from the direct action of the flames.

We have no knowledge of any steel casting firm using an instrument for determining the temperature in their annealing furnaces, it being judged entirely by the eye. With long experience this method is doubtless more or less accurate, some experts claiming to be able to read to 15° F. When those of less experience have control of the annealing, it seems but reasonable to insist on the use of the pyrometer, while at the same time we recognize the fact that the users of steel castings have to do with the physical qualities rather than with the methods or details employed in obtaining them.

In the annealing of forged steel ordnance by the leading manufacturers in this country, experts are in charge who at the same time have at their command the most improved instruments. The known results obtained speak for themselves. If by chance the temperature has reached too high a point, the piece is cooled and re-heated, showing that the beneficial changes take place as the temperature rises.

GENERAL REMARKS.

The following may be noted in connection with Passed Assistant Engineer Reynold T. Hall's article on "Steel Castings," August JOURNAL, 1889, showing by comparison the more recent practice.

One half inch to three inches of facing sand is found sufficient for castings of the various sizes and weights.

There is comparatively little nailing done. The presence of the nails is not essential to the stability of the facing, and their absence gives a greater area for the escape of air and gases.

The venting is by the use of "pops," one quarter inch to one half inch holes through the cope.

The shrinkage in small sections is not greater than in larger ones, the disadvantage of light webs connecting heavy flanges, as in some bed plates, is that such defects as porosity and checks are likely to develop.

It is believed that heads influence more the soundness of the castings than density of the metal.

Cinders are used in the molds and cores not only because of their openness, but that they offer less resistance to the demands of shrinkage.

Ladles are heated by gas entering at top or bottom, or both.

A greater variety of and purer stock is used.

It is seldom necessary to loosen the flask fastenings to accommodate the casting while shrinking.

Ferro manganese is added about five or ten minutes before tapping.

Steps are not taken to compel a casting to shrink in any particular direction; the endeavor being so to construct the molds and cores that the casting may move at will.

Rapid cooling is not resorted to.

Unless the carbon is high, annealing with iron oxide at a cherry red temperature does not appreciably affect it.

No difference has been found in the specific gravity of a steel casting at its upper and lower extremities.

During the past year the prices of steel castings have declined about 25 per cent.

From Assistant Engineer C. A. Carr's article on the "Failure of Steel Castings," February JOURNAL, 1892, we learn that the experience with steel castings for the *Maine* and *Cruiser No. 11*, of the Quintard Iron Works of New York, and the Government inspectors at the company's works, was exceedingly annoying, discouraging and expensive, and calculated to give rise to a feeling of distrust if not prejudice. The Quintard Iron Works

were not alone in their misfortune, as Mr. Carr but too truthfully states, but suffered in common with other builders of Government machinery.

Referring to the tables showing the castings of steel intended to be used in the construction of the machinery of these vessels, it is found that, with the exception of the engine columns and frames, no very material departure was made from the original intention; but when we are informed of the unfortunate circumstances, the repeated failure of castings, of delay in delivery, and accompanying expenses, which led to the changes, it presents itself in quite a different light, and causes but little surprise to learn that the builders were not disposed to machine a steel casting, even though in the rough it was to all appearances sound, if permitted to substitute some other metal, their previous experience causing them to feel that it would simply result in loss of time and money.

In reviewing the castings rejected, and comparing them with recent productions, we feel warranted in expressing the belief that should similar castings be required to-day, there would be no difficulty experienced in furnishing them, and so much improved in every particular that the builders could not but feel satisfied, and encouraged to extend the use of steel castings.

The columns for the *Maine* for which forged steel was substituted, could, with a slight modification of core and intelligent supervision, be made without difficulty.

It will be noticed that the seams or checks existed, with but two exceptions, in the vicinity of the bosses shown in Fig. 12. The cores leave considerable masses of metal at *b b*. When poured, this metal did not set until after that of less cross section immediately adjacent, and, consequently, there was, as in all similar cases, a struggle to maintain the continuity of the castings into which the molds, the cores and the metal entered. A change in the form of core, making it concentric with outside of bosses, as shown in dotted line at *d d*, would have equalized the cross section and lessened the strife within the metal, and by relieving the castings of the restraint of the molds soon after pouring, thus permitting them to creep and accommodate them-

selves to the demands of contraction, the castings would, in all probability, have been sound. Had not the molds held the castings fast while the metal was setting, the longitudinal shrinkage would doubtless have been about three-sixteenths inch to the foot instead of *nil.*

These columns afford an excellent and simple illustration of the effect of flanges, or any projection that may prevent a casting from moving in the mold while the metal is setting, a most fruitful source of shrinkage cracks, and aided, in this instance, by the difference in cross sections, as mentioned.

When broken up for scrap, many of the checks mentioned by Mr. Carr were found to extend entirely through the castings, and, in some cases, where the defects on the surface were the least noticeable, they developed into decided cracks immediately underneath.

Of all the castings the writer has seen broken into scrap, these, without exception, were the worst, and their condition was undoubtedly very much more the result of ignorance than of fault of design.

His attention having been called to the pistons referred to on their return to the steel works, he is prepared to say that, with one exception, the defects mentioned were shrinkage cavities of no considerable depth, though amply sufficient to warrant the rejection of the castings. No imperfections were found in any other part of the castings, which was learned by breaking them up for the purpose of exploration and remelting.

In replacing the piston rejected owing to deep holes at inner edge of follower seat, a good sized fillet resulted in a sound casting.

In the exception mentioned above, though the surface defects were of about the same extent as in the others, they led to large internal shrinkage cavities and a crack of several inches in length, extending nearly to the surface on both faces of piston.

What has been said concerning the pistons, other than the exception, applies also to the crossheads.

The bed-plate for *Cruiser No. 11*, rejected on account of cracks on each side of crank-shaft bearings, was broken up for scrap,

and the cracks found to disappear about three-fourths inch below the machine surface.

In the casting of the Y frames for *Cruisers Nos. 9, 10* and *11* much difficulty was experienced, and, on application, the Engineer-in-Chief of the Navy authorized the changes suggested by the manufacturer. They were not approved as a design, but rather as a means to an end, and that they might be cast of open-hearth steel, and also (as in many other similar cases of changes or modifications of designs, or acceptance of castings) to facilitate the building of the new vessels, and the encouragement of the manufacture of steel castings; this last fact, the firms admit, has characterized the Bureau of Steam Engineering in all its relations with them.

The changes suggested and approved were of the most radical nature, dividing each frame into three sections with flanges—two legs and an upper piece—the three bolted together to form the Y, as shown by Mr. Carr. The result of the changes was that the frames were finally cast, but with difficulty, and used for *Cruisers 9* and *10*. Should similar castings to the original design be required at the present time, we feel confident that no such alterations would be necessary.

It is a most common error to speak of certain irregular holes and cavities found in some steel castings, as blow holes, when, as a matter of fact, a large percentage of them are caused by shrinkage.

It is not unusual for shrinkage cavities and small holes to show where heads or runners joined the casting; they are generally local and may be explored to their fullest extent, and in many instances will machine out, at all events reduce in size as the machining progresses.

In a blowy casting there is usually no improvement by machining; in fact, the tendency is to grow worse during the operation.

In the rock-shaft arm if the hubs are not efficiently headed, the shrinkage, progressing from the outer surfaces, draws on the adjacent molten metal, and finally leaves a space unsupplied where the metal remained liquid longest. After boring out the hubs, the cutting for keyways will probably lead into shrinkage

cavities on the sides nearest the arms, but on machining a little farther into the castings, solid metal would likely be reached. While the fact remains that the presence of any such defect at the point named does of necessity condemn the casting for the purpose intended, the object here is simply to draw attention to the frequent presence of shrinkage holes under certain conditions of heading.

The closed crack in engine column for the *Maine*, mentioned by Mr. Carr, justly implies irregularities on the part of some one; and this is not the only instance of the kind to which the writer's attention has been called, but it is extremely gratifying to be able to say, and in justice to the firms with which he has been associated, that there has been no attempt whatever to conceal any defects that may have appeared in castings presented for inspection, the disposition being decidedly the reverse, and attention called to such defects.

To the writer's knowledge, the most emphatic orders have been given to subordinates and workmen to avoid even accidentally anything of the kind. It is not surprising, however, that now and then an employé may be found who, from a desire to benefit or injure his employer, might be led to an act of this kind.

An inspector may easily satisfy himself in this particular by keeping the castings in view from the time they are out of the sand until officially presented for inspection and shipment.

The following is offered for comparison:

The weight of steel castings shipped for armored and protected cruisers, and the battle ships now under construction, has been 845 tons; of this there was rejected at the works of the builders 13.67 tons, or about 1.6 per cent.

Shipped for harbor defence *Ram*, 29 tons, and rejected 179 pounds, or about .3 per cent.

From Mr. Carr we have for the *Maine*, shipped 52 tons, and rejected 16 tons, or about 30.7 per cent. on the same basis.

The percentage of rejection for the other vessels of the Navy building at the same time as the *Maine* was doubtless nearly the same.

The marked difference in the percentages rejected in the favor of present results will be observed.

TEST SPECIMENS.

Concerning material to be taken for specimens, a portion of Par. No. 131, *Torpedo Boat No. 2*, Dec. 1st, 1892, reads as follows: "Be cut either from coupons to be molded and cast on to some portion of the casting, or from sinking heads in cases where such heads of sufficient size are employed. Coupons to be so fixed as not to interfere with the successful making of the casting, but at the same time showing the quality of the material."

The custom is to use the coupons for specimens, though very frequently they do not represent "the quality of the material" being much inferior, for very good reason.

The form of coupon is a parallelopipedon, 4 inches $\times$ 4 inches + 13 inches for 8-inch specimens. When a chunk of this size is attached to a 1-inch or 1½-inch flange the size of the neck joining the two is necessarily limited, the result being that the neck loses its fluidity before the coupon, and in some cases before that particular part of the casting, the supply to the coupon is therefore cut off, leaving it unsound from the presence of shrinkage cavities. Again, if the coupon mold is not sufficiently open, the air and gases which cannot escape through this medium, and the gases evolved by the solidifying metal, are confined, and the coupon, in addition to containing shrinkage holes, is blowy, while the casting which it represents may be free from all of these defects. In no case when specimens have been taken from the body of a casting, owing to the impossibility to procure sound ones from its coupon, have they been imperfect. In some instances the specimens were cut from the flange to which the coupon had been attached, and in others from the body proper of the casting.

With uniformly satisfactory results a flange or some portion of the casting has been extended to be used for specimens, but it is not every shape that will admit of this without considerable expense in cutting off the extension; then, again, the size of the flask may not leave available space.

Table I.

EIGHT-INCH SPECIMENS.—MATERIAL ANNEALED.

Ult. ten. strength.	Elastic limit.	Elon. in 8 inches.	R'd'ction of area.	Carbon.	Manganese.	Silicon.	Sulphur.	Phosphorous.	1 in. sq. bars bent over a radius of 1½ times thick.
lbs. p. sq. in	*lbs. p. sq. in*	*per cent.*	*per cent.*	*p. c't.*	*p. c't.*	*p. c't.*	*per c't.*	*per c't.*	
61,200	27,200	24.5	32.	.24	.53	.20	.027	.044	180°
61,400	27,200	25	35.67	.24	.53	.20	.027	.044	
62,000	27,200	23.12	29.5	.24	.53	.20	.027	.044	
63,910	29,740	23	39.31	.26	.57	.27	.06	.059	140°
63,600	29 600	25	40.59	.26	.57	.27	.06	.059	
64,260	30,220	19.12	22.25	.26	.57	.27	.06	.059	
65,290	30,580	19.12	24.95	.26	.57	.27	.06	.059	
62,000	24,000	20	23	.26	.69	.228	.043	.042	180°
62,400	24,000	27.88	39.63	.26	.69	.228	.043	.042	
65,200	26,000	16.25	17.67	.22	.65	.256	.042	.036	
65,200	26,000	16	16.32	.22	.65	.256	.042	.036	
66,200	26,600	21	24	.24	.52	.20	.037	.048	140°
67,800	26,600	21	24	.24	.52	.20	.037	.048	
67,000	30,600	24.63	36.47	.28	.63	.177	.037	.044	160°
67,600	30,200	24	33.65	.28	.63	.177	.037	.044	
70,600	32,800	17.23	19.68	.32	.6	.233	.051	.06	125°
68,600	32 800	19.75	26.20	.32	.6	.233	.051	.06	
71,200	32,600	18.13	15.86	.20	.46	.22	.037	.057	120°
71,600	32,800	18.75	17.68	.20	.46	.22	.037	.057	
71,400	28,600	20.5	26.51	.25	.68	.239	.049	.048	150°
73,000	28,800	19.75	18 57	.25	.68	.239	.049	.048	
63,200	36,000	14.12	18.58	.21	.503	.221	.034	.083	45° to 135°
69,800	35,200	15	14.7	.21	.503	.221	.034	.083	
64,800	36,800	17.5	21.93	.19	.5	.217	.034	.084	
66,000	36,600	19	23.27	.19	.5	.217	.034	.084	
66,200	37,600	24	40.99	.21	.56	.22	.034	.081	
64,800	37,400	15.63	16.31	.21	.56	.22	.034	.081	

Table II.

EIGHT-INCH SPECIMENS.—MATERIAL NOT ANNEALED.—TWELVE HOURS IN SAND.

68,000	...	15.63	17.21	.23	.65	.226	.05	.044	60° to 90°
72,400	...	14.13	11.16	.22	.61	.214	.046	.045	
74.800	...	15.63	15.16	.23	.66	.226	.041	.047	
69,200	...	19.31	23.49	.23	.63	.226	.041	.041	
76.400	...	12.5	14	.30	.63	205	.039	.047	
70,000	...	19.25	18.8	.23	.60	.221	042	.04	
70,400	...	18	20.13	.25	.55	.183	.052	.042	
74,400	..	17.25	22.38	.25	.60	.233	.045	.042	

The coupons are usually joined to the casting by one neck, as shown in Figs. 9 and 10, the metal entering in the direction shown by the arrows. A method which rids the coupon of the air and gases which may have been in the molds is shown in Fig. 11, and with this method, if attached to a moderately heavy section, the coupon will fairly well represent the casting.

Frequently castings have been lost by shrinkage caused by the presence of a coupon. Gates* and heads will have the same effect if not intelligently arranged.

The foregoing tables, showing results of tests of coupons from steel castings are given to show the characteristics of the metal as now made. They have been selected at random, and may be considered as average figures.

In closing this paper, the writer desires to express his appreciation of the constant courtesy shown him by the officials and employés of the steel casting works where he has been stationed as inspector, the Standard Steel Casting Company, of Thurlow, Pa., and the Norristown Steel Company, of Norristown, Pa. As is well known, the former have furnished most of the steel castings for our new ships, and most of the writer's time was spent there. Every facility for inspection at every stage of the work was afforded at both places, and, as has already been stated, the aim seemed to be to call attention to defects, when any occurred, rather than to attempt any concealment.

*The term gate is used to denote the channel through which the metal enters the mold.

ECONOMICAL SPEED AND COAL ENDURANCE OF WAR VESSELS, AS AFFECTED BY THE RELATION OF THE COAL EXPENDED FOR PROPULSION TO THAT EXPENDED FOR ALL OTHER PURPOSES.

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The coal expenditure on board ship is divided up among so many engines, auxiliaries and losses that the steaming qualities of a new ship can be determined only by trial at sea on long cruises, and then only as an approximation to the truth. After a year's commission the endurance can be worked out from the steaming log for a few speeds, and the amount of coal required to propel the ship one knot determined with fair accuracy; but nothing is actually gained to enable a designer to predict the results beforehand, even for a ship of the same class. It is a common experience to find one cruiser more economical than another without being able to state the reason or put one's finger on the actual point of gain. (1) It may be the size and height of the smoke pipe which produces more complete combustion with a minimum quantity of coal. (2) It may be a better design of boilers which evaporate more water for one pound of coal; (3) it may be better lagging which prevents radiation from the boilers and pipes, and less drainage leak; (4) it may be greater efficiency of auxiliaries run with the main engines; (5) it may be smaller expenditures for auxiliary purposes outside of the machinery department; (6) it may be due to better fitting of packing rings, and, finally, (7) it may be superiority of design of main engines or screw.

There are so many separate elements about which no exact data have been obtained on the modern man-of-war that the record of the performance as a whole becomes a rough approximation, which cannot be used except in the most general way.

We know well enough that a saving of coal will increase the steaming radius, but we neither know how much, nor how it affects the speed at which the greatest distance can be made. While the design of the propelling engines has been reduced to a scientific method, our treatment of the auxiliaries is still far from satisfactory. We have assumed that the large expenditures for them are inevitable and cannot be reduced enough to make it worth while to carry out systematic investigations for the purpose of determining exactly what saving might be effected.

The auxiliaries on board ships may be divided into three classes: (1) Those which are run only when the main engines are run, such as the air pumps, circulating pumps, feed pumps and, at high speeds, the forced-draft blowers; (2) those which are run steadily for long periods in port as well as at sea, such as bilge pumps, ventilating fans, dynamos, distillers and evaporators; (3) those which are run from time to time, as occasion requires, such as boat winches, anchor windlass, turret-turning machinery and fire-pumps, or only at times when the coal expenditure is of no importance, such as auxiliary air and circulating pumps and auxiliary boiler feed pump. To the above must be added losses from condensation in the pipes and boilers, radiation and leakage through joints and valves; also, expenditure for galley and heaters.

The horse power and coal expenditure for the first class, which I shall call *main engine auxiliaries*, has a certain initial value at the starting of the main engines, and then increases at a rate dependent upon the power of the main engines. As may be seen in tables published in the last issue of the JOURNAL, their power is nearly constant at low speeds of the ship and even up to a speed somewhat above the economical rate.

The auxiliaries of the second class, which may be called *general*, or *ship's auxiliaries*, require practically a constant expenditure of coal at all speeds of the ship.

The auxiliaries of the third class, which may be called *special auxiliaries*, are of no importance here, since we are considering only the question of obtaining the greatest economy for cruising speeds.

The losses, under which head it may be well to place the galley and heaters, although the last named are on only when the ship is in a cold climate, must be made up by coal expenditure, which is fairly measured by the amount required to keep steam under banked fires with no machinery running. The loss from radiation and condensation is proportional to the number of boilers and the length of steam piping in use. The loss up the smoke-pipe will not be considered in the following. Each engine bears its own share of that in the allowance of water evaporated per pound of coal.

A list of all the machines on board the armored cruiser *New York* is added for the purpose of classifying the auxiliaries.

	Steam cylinders.
MAIN ENGINES	
4 propelling engines, vertical, triple-expansion	12
MAIN ENGINE AUXILIARIES.	
4 main air pumps, Blake, vertical, duplex	8
4 main circulating pumps, horizoutal, single	4
3 feed pumps, Blake, vertical, duplex	6
3 auxiliary feed pumps, Blake, vertical, duplex	6
12 forced-draft blowers, vertical, double	24
GENERAL AUXILIARIES.	
2 fire and bilge pumps, Blake, vertical, duplex	4
2 ventilating fans for engine rooms, vertical, double	4
4 ventilating fans for ship	8
3 dynamo engines, vertical, double	6
1 steam steering engine, Williamson	2
4 evaporator and distiller pumps, Davidson	5
1 ice machiue—Allen Dense Air	1
SPECIAL AUXILIARIES.	
2 auxiliary feed and fire pumps, Blake, vertical, duplex	4
4 main and auxiliary fire pumps, Blake, vertical, duplex	8
2 auxiliary boiler feed pumps, Blake, vertical, duplex	4
2 wrecking pumps, Blake, horizontal	2
2 water service pumps, Blake, vertical, duplex	4
2 auxiliary air and circulating pumps, Wheeler, horizontal	2
1 anchor windlass, American Ship Windlass Co	2
6 steam winches, Williamson	12
2 auxiliary boiler blowers, vertical, double	4
6 ash hoisting engines, vertical, double	12
2 turning engines, vertical, double	4
4 reversing engines, inclined, single	4

	Steam cylinders.
2 turret turning engines, horizontal, double	4
2 ammunition hoists, vertical, double	4
1 workshop engine	1
1 air pump for sweeping decks, Westinghouse	1
STEAM LAUNCH ENGINE.	
2 propelling engines, vertical, compound	4
2 feed pumps, duplex	4
92 engines	170

Careful experiment on board of the new ships will be necessary to determine all the elements which affect the speed, but, for the present, the accompanying Table 1 of the *Charleston's* economic qualities will serve to indicate, in a general way, the expenditures for the different purposes referred to above. The horse powers and coal set down in the columns are only fair approximations, as no experimental trials have ever been made to fix these quantities for different rates of speed. There is no doubt, however, that the general results are accurate enough to point out the reason for the great falling off in economy at low powers of the main engines—that is, the economy for propulsion alone. It will be seen that 74.2 per cent. of the coal is expended on auxiliaries and losses at 4 knots, 46.6 per cent. at 8 knots, and 15.8 per cent. at 18 knots. The coal for the main engines is supposed to include all that which goes into the main cylinders for useful work—friction of the moving parts and the screw, churning of the screw, condensation within the cylinders, and radiation from the engines alone.

While a continued study of the main engines develops many improvements for increasing their economy, the gain is relatively small at low powers, where the coal expenditure becomes so important a question for war ships. It is rather in the auxiliaries and losses that the greatest opportunities for increasing endurance are to be found.

Take the problem which the Naval Engineer has set before him, to produce machinery that will be economical at low powers, and yet shall have strength and steam capacity for enormous powers and very high speeds, the latter to be sustained for long periods, if necessary. All this must be accomplished with low

Table I.

SHOWING COAL EXPENDITURE OF U. S. S. *CHARLESTON*.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Knots speed.	I.H.P. main engine.	Coal per I.H.P.	I.H.P. air pump.	I.H.P. circ. pump.	I.H.P. feed pump.	I.H.P. blower.	I.H.P. main bilge pump.	I.H.P. other auxiliaries.	I.H.P. total auxiliaries.	Coal per I.H.P. aux's.	Coal per hour, aux's.	Coal for radiation, leakage and losses.	Coal for distilling, etc.	Coal for main engines.	Total coal per hour.	Knots per ton of coal.	Per cent. coal on main engines.
4	107	4.0	23.7	19.4	2.5	0	6.9	26.5	79.0	6	474	655	100	428	1657	5.41	25.8
5	190	3.5	23.9	20.1	2.8	0	6.9	26.5	80.2	6	481	655	100	630	1866	6.00	33.8
6	270	3.25	24.2	20.8	3.1	0	6.9	26.5	81.5	6	489	655	100	878	2122	6.33	41.4
7	380	3.0	24.6	21.7	3.6	0	6.9	26.5	83.3	6	499	655	100	1140	2394	6.55	47.6
8	520	2.8	25.1	23.2	4.2	0	6.9	26.5	85.9	6	515	655	100	1456	2726	6.57	53.4
9	700	2.6	25.7	24.6	5.0	0	6.9	26.5	88.7	6	532	655	100	1820	3107	6.48	58.6
10	960	2.5	26.5	27.3	6.2	0	6.9	26.5	93.4	6	560	780	100	2400	3840	5.83	62.5
11	1290	2.4	27.6	30.6	7.5	0	6.9	26.5	99.1	6	595	780	100	3096	4571	5.39	67.7
12	1710	2.3	29.0	35.9	9.3	0	6.9	26.5	107.6	6	646	905	100	3933	5584	4.82	70.4
13	2220	2.2	30.7	41.1	11.4	0	6.9	26.5	116.6	6	700	1030	100	4884	6714	4.34	72.7
14	2820	2.1	32.7	47.5	14.0	2.5	6.9	26.5	130.1	6	781	1030	100	5922	7833	4.00	75.6
15	3550	2.0	35.2	58.0	17.1	6.4	6.9	26.5	150.1	6	901	1030	100	7100	9131	3.68	77.8
16	4370	1.9	37.9	70.0	20.6	17.6	6.9	26.5	179.5	6	1077	1030	100	8303	10510	3.41	79.0
17	5220	2.1	40.7	84.8	24.0	36.8	6.9	26.5	219.7	6	1318	1030	100	10962	13410	2.84	81.8
18	6120	2.5	43.7	114.0	28.0	69.6	6.9	26.5	288.7	6	1732	1030	100	15300	18162	2.22	84.2

weight. The problem is like building up a working man who shall have the appetite of an infant when he has little work to do, and yet who retains the strength and stomach capacity to work and eat like a giant in flush times. To some extent, men do eat in proportion to the expenditure of muscular energy, but there is a certain minimum food supply required to keep the body up to its normal condition, whether they work or not, and this is true of marine engines and boilers. A certain amount of coal has to be burnt before the ship is in a condition to move at all, and it is this which forms so important a part of the expense on a man-of-war. Take the case given in the table. The power expended on the main engine auxiliaries does not vary much below 10 knots, and it probably never gets below a value of 45.6 I.H.P., even when the ship is lying under banked fires. The power required for the ship's auxiliaries is practically constant at all speeds. Its value is 33.4 I.H.P., which may be added to the above, giving 79 I.H.P., and (at 6 pounds of coal per I.H.P.) 474 pounds of coal per hour for the auxiliaries before the main engines can move at all. The allowance of 6 pounds of coal per I.H.P. in this case is fully justified by calculations of steam expenditures from the terminal pressures of the indicator cards taken on the full-power trial trip. As the auxiliaries are all driven by simple engines, with little or no expansion, and were run at high powers, the above probably represents the minimum expenditure.

It will not be necessary to state here the method of obtaining the powers and coal for low rates of running, as a short explanatory paper on that subject was published in the November number of the JOURNAL. The percentage column shows at once how important the design of small things becomes, and how close should be the attention to coal expenditures, outside of the engine room.

The most economical speed is often discussed, as if it had a fixed value for a given ship. Many conflicting reports on this economical speed have been received from our new ships, and the data taken from the steaming log are not very satisfactory.

As a matter of fact, the economical speed is not a constant,

but may vary two or three knots in the same ship under different conditions of steaming. Generally speaking, it is dependent on the coal expended for propulsion and that expended for all auxiliary purposes, and, as will be shown later, its value is fixed by the proportion which these two bear to each other. All other elements which affect the speed may be reduced to a function of these.

Suppose, for the moment, that no coal is expended for auxiliary purposes, losses, &c., and that every pound of coal goes into propulsion alone; then the economical speed will fall to a very low rate, limited only by the slip of the screw. Two knots would be nearer the most economical speed in such a case. As stated above, the coal expended outside of the main engines never gets below a certain constant quantity per day for a given ship. This is exactly like a current which the ship has to overcome before she can move from her anchorage; the greater the current the higher the speed at which the ship must steam through the water to get over the ground at all, and, it naturally follows, the higher will be her most economical speed. Graduates of the Naval Academy will recall the old problem of determining the economical speed of a ship steaming against a current. It bore a constant ratio to the current which, as I recall it, was $1\frac{1}{2}$ times the current velocity. Exactly the same thing is true of the coal expense for auxiliary purposes, the greater it is in comparison with the coal for propulsion, the higher the economical speed, and conversely, the less its proportion to the coal for propulsion, the lower the economical speed. A slight consideration of the subject will show that any decrease of coal expended on propulsion increases the proportion borne by the auxiliaries, and consequently raises the economical speed.

The analogy between the expense for auxiliaries and a current is not perfect, as the former is added to the coal per day, while the latter is subtracted from the speed, but it serves to illustrate the point well enough.

In general it may be stated that any change in the proportion which the coal for auxiliaries bears to that for propulsion produces a like speed in the economical change. This may be demonstrated mathematically as in the following:

[The following expression for the economical speed was worked out by Chief Engineer John Lowe, U. S. N., ten years ago.]

We assume first that the coal burnt per hour is equal to a constant plus a quantity which varies as the cube of the speed. While this assumption is not strictly correct, it will be found substantially so within the limits of ordinary cruising, from 5 to 16 knots for the cruisers. Above 16 knots the resistance to the ship's motion varies at a higher power than the square of the speed, while below 5 knots, the slip and resistance both depart considerably from the mean law. A table of comparison between the *Charleston's* actual expenditure and that calculated from the above is shown below. The error is seen to be small.

Let S = speed of ship, H = the corresponding horse-power for propulsion, D = displacement of ship, and C = a constant; then $H = \frac{D^{\frac{2}{3}}S^3}{C}$, at moderate speeds. Let a be the number of pounds of coal per I.H.P. per hour for propulsion, assumed to be constant.

The coal burnt per hour for propulsion will be

$$\frac{aD^{\frac{2}{3}}S^3}{C} = N_1S^3, \text{ taking } \frac{aD^{\frac{2}{3}}}{C} = N_1.$$

Let N_2 = the number of pounds of coal burnt per hour for all purposes outside of propulsion, assumed to be constant at the speeds considered, and T = the total weight (in pounds) of coal burnt on the grates per hour.

$$\text{Then } T = N_1S^3 + N_2,$$

$$\text{and } E = \frac{2240\ S}{N_1S^3 + N_2} = \text{knots steamed for one ton of coal.}$$

Differentiating to find the most economical speed, we have—

$$\frac{dE}{dS} = \frac{(N_1S^3 + N_2) - 3N_1S^3}{(N_1S^3 + N_2)^2} = 0$$

$$= \frac{N_2 - 2N_1S^3}{(N_1S^3 + N_2)^2} = 0$$

from which (1) $$S = \sqrt[3]{\frac{N_2}{2N_1}}$$

$$(2) \qquad E = \frac{2240\sqrt[3]{\frac{N_2}{2N_1}}}{N_1\left(\frac{N_2}{2N_1}\right) + N_2} = \frac{1493}{N_2}\sqrt[3]{\frac{N_2}{2N_1}} = 1493\sqrt[3]{\frac{1}{2N_1N^2_2}}$$

Equation 1 gives the most economical speed, and equation 2 the distance steamed by the ship on 1 ton of coal at that speed.

It will be seen that S falls for a decrease of N_2, the coal for auxiliaries, and rises for a decrease of N_1, the coal for propulsion; also, that E increases for both cases.

In other words, a saving of coal and an increase of the steaming radius changes the economical speed either up or down, according to the place where the saving is effected.

The accompanying table has been made out for the *Charleston* from the formula $T = N_1S^3 + N_2$, in which $N_1 = 2$, and $N_2 = 1700$. The coal, as set down in the JOURNAL for November, is added; also a percentage column, corrected to agree with the formula.

Table II.

Speed in knots.	Coal per hour from formula $T = N^1 S^3 + N_2$.	Coal per hour from indicator cards.	Per cent. coal expended on propulsion from formula.
4	1,828	1,657	7.0
5	1,950	1,866	12.9
6	2,132	2,122	20.3
7	2,386	2,394	28.8
8	2,724	2,726	37.6
9	3,158	3,107	46.2
10	3,825	3,840	54.1
11	4,487	4,571	61.0
12	5,406	5,584	67.0
13	6,469	6,714	72.1
14	7,563	7,833	76.3
15	8,825	9,131	79.9
16	10,267	10,510	82.8
17	11,901	13,410	85.2
18	13,739	18,162	87.3

The percentage of coal expended on propulsion excludes even the friction of the main engine, and drop and condensation

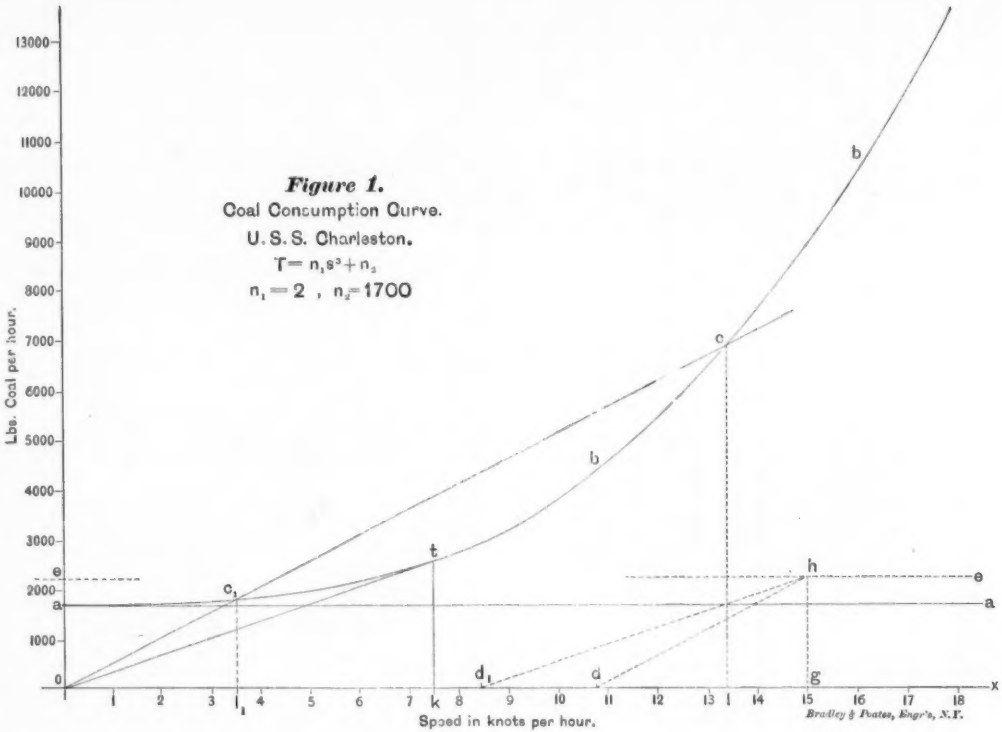


Figure 1.

Coal Consumption Curve.

U. S. S. Charleston.

$T = n_1 s^3 + n_2$

$n_1 = 2$, $n_2 = 1700$

within the cylinders. It is, therefore, much lower at low speeds than the percentage given to the main engines in the first table.

The formula $T = N_1 S^3 + N_2$ may be represented graphically as in Fig. 1, and the effect of a change in the coal consumption upon the speed may thus be studied from a coal consumption curve. The curve for the *Charleston* is given in Fig. 1. OX is the base line upon which the speed is laid off; $Oa = 1700$, the constant weight of coal laid off vertically. The curve *abb* represents the variable weight $N_1 S^3$, laid off from a—a. A similar curve can be constructed for any ship by weighing the coal expended for a series of runs at different speeds, and such a curve would be far more useful to officers who go to sea in the ship than the ordinary speed and power curve.

The distance steamed for one ton of coal may be obtained at once from this curve. Take the speed ol, for instance, and draw the vertical lc cutting the curve; also draw Oc. Lay off $gh = 2240$ pounds, and draw eh parallel to OX. The line hd parallel to Oc will cut off gd equal to the distance steamed for one ton of coal. This is evident from the similarity of the triangles, Ocl and dhg, $cl : ol :: hg : gd$. $cl = T$, $ol = s$, $hg = 2240$ and $T : s :: 2240 : gd$,

$$gd = \frac{2240 \times s}{T} = \left(\frac{s}{\frac{T}{2240}}\right).$$

The value of gd is obtained for any other speed by a similar construction. In the equation, $gd = \frac{2{,}240 \times s}{T}$, the quantity $\frac{s}{T}$ is the tangent of the angle which Oc makes with a vertical to OX, and this is evidently a maximum when the line drawn from O is tangent to the curve; hence a perpendicular from the point of tangency t determines the speed OK of maximum endurance for one ton of coal, or the most economical speed. The line hd_1 drawn parallel to Ot determines gd_1, the maximum endurance for one ton of coal. At the speed Ol_1, determined by the point c_1, where Oc cuts the curve a second time, the ship has the same endurance as at the speed Ol. Since any line through O, above the limit Ol, must cut the curve in two points, there are always two speeds giving the same steaming radius.

In Fig. 2, we see the effect of a change in the coal consumption. Suppose the coal for propulsion be increased by reason of foul bottom, deeper draught, head wind and sea, or greater wastefulness in the cylinders ; then *Acc* represents the curve of coal consumption, and the speed *Ol*, determined by the tangent *Or*, is less than *Ok*. If, on the other hand, the coal for propulsion be decreased, and *Add* represent the curve, Or_1 becomes the tangent, and Ol_1 the economical speed which is greater than *Ok*.

If the coal for auxiliaries be increased from *Oa* to O_1a the tangent O_1t_1 to *abb* gives Ok_1 as the economical speed, greater than *Ok*. If O_1a had been less than *Oa*, this speed would have been less than *Ok*.

The exact change in the speed and endurance may be determined from the formulæ

$$S = \sqrt[3]{\frac{N_2}{2\,N_1}} \text{ and } E = 1493\sqrt[3]{\frac{1}{2\,N_1 N^2_2}}.$$

Let N_1 for propulsion be decreased or increased p per cent., then s will be increased or decreased $\frac{p}{3}$ per cent.

Let N_2 be increased or decreased p per cent., then s will be increased or decreased $\frac{p}{3}$ per cent.

If both be increased or decreased alike, s does not change.

Since E varies inversely as $\sqrt[3]{N_1}$ and $\sqrt[3]{N^2_2}$, any decrease in either increases the endurance.

A decrease of p per cent. for propulsion increases the steaming radius $\frac{p}{3}$ per cent., while a decrease of p per cent. for auxiliary purposes increases the steaming radius $\frac{2\,p}{3}$ per cent., showing that the gain by saving on the auxiliaries is just twice as great as by saving on propulsion. Let it be remembered, too, that the same percentage of gain in both cases is not accompanied by any sacrifice of speed.

The above formulæ are given as guides in questions affecting a ship's steaming at sea to enable an engineer to ascertain the

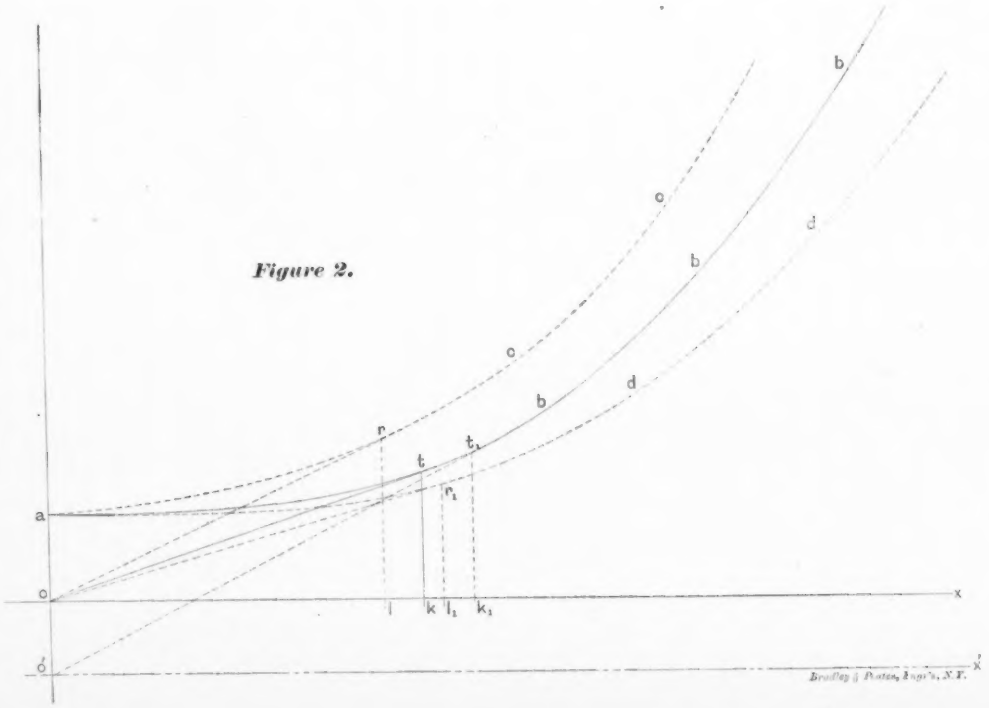


Figure 2.

best rate of speed for different conditions of the sea, weather, bottom and machinery, and not as quantative or even final. As an illustration I add the following:

The *San Francisco's* economical speed is about 10.5 knots and her endurance 4,259 knots. Suppose that a strong head wind or foul bottom should double her resistance, then N_1 would be increased 100 per cent. and the economical speed would fall to 8.3 knots and the endurance to 3,380 knots.

Suppose that by shutting down dynamos, distillers and all outside expenditures, the coal for auxiliaries could be reduced 40 per cent., then the economical speed would fall still further to about 7.3 knots while the endurance would rise to 4,259 knots again.

The results demonstrated by these formulæ are only what might have been expected. Naval engineers know well the ruinous expense of separate auxiliaries at low powers of the main engines, and, also, the great expense of providing for keeping the boilers in readiness for steaming at any speed.

I have had occasion to say that triple-expansion engines are not economical for cruisers running at moderate speeds, and it may be worth while to show in a table the best possible results that could be obtained from ideal engines of the different types. The accompanying table No. 3 is worked out for that purpose, taking 100 cubic feet of steam per minute as a basis of calculation and supposing the cylinders to be perfect non-conductors, the steam to be dry but not superheated, and to be supplied with heat enough to keep it dry during expansion, also the cut offs to be so adjusted that there is no drop in the receivers with no wire drawing from the boiler to the cylinders. All of the conditions are ideal, and give the highest results attainable with a perfect engine. The four grades of expansion taken correspond nearly to those common with marine engines of the four types—simple, compound, triple-expansion and quadruple-expansion. The steam pressures are those commonly used. The back pressure in the L.P. cylinder is taken at 3 pounds per square inch.

Table III.

SHOWING STEAM USED PER I.H.P. PER HOUR AT DIFFERENT GRADES OF EXPANSION FOR PERFECT ENGINE.

Steam pres. abs.	Weight of steam per hour.	Simple engine. Expansions=3		Compound engine. Expansions=5		Triple expansion. Expansions=8		Quadruple expansion. Expansions=12	
		I.H.P.	Steam per I.H.P. per hour.	I.H.P.	Steam per I.H.P. per hour.	I.H.P.	Steam per I.H.P. per hour.	I.H.P.	Steam per I.H.P. per hour.
200	2615.4	179.2	14.60	221.2	11.82	258.3	10.12	**288.5**	**9.06**
160	2118.0	142.6	14.85	175.6	12.06	**204.6**	**10.35**	227.8	9.30
135	1805.4	119.7	15.08	147.3	12.26	171.1	10.55	189.6	9.52
110	1490.4	96.7	15.41	118.7	12.56	137.2	10.86	151.3	9.85
90	1234.8	78.5	15.73	**96.0**	**12.97**	110.3	11.19	121.0	10.20
75	1041.6	**65.0**	**16.06**	78.8	13.22	90.4	11.52	98.4	10.59
50	712.8	41.9	17.01	50.4	14.14	56.9	12.53	60.2	11.84
25	372.0	19.0	19.58	22.0	16.91	23.0	16.17	22.5	16.53

Back pressure in L. P. cylinder = 3 lbs.
Steam used per minute = 100 cu. ft.

The horse powers are worked out from the formula, I.H.P. = $PV \div 33{,}000$.

The foot pounds of work = PV, in which V = volume of steam at time of exhaust in cubic feet = 100 r, and r = number of expansions.

Let P = mean pressure per square foot.

p = initial pressure per square inch.

$$\text{and } P = 144\left(p\left(\frac{1+\log_e r}{r}\right)-3\right) \text{ so that}$$

$$\text{I.H.P.} = \frac{14400\, r\left(p\left(\frac{1+\log_e r}{r}\right)-3\right)}{33000}$$

The weight of steam is obtained from Peabody's tables. As will be seen, the theoretical gain in going from simple to compound engines was 19 per cent., from compound to triple-expansion engines, 20 per cent., from triple-expansion to quadruple-expansion will be about 12·5 per cent. The above is purely theoretical, and supposes the engines to be worked up to their highest

designed power. Every engineer knows that such results can never be obtained in actual practice. They may be approached where engines are run steadily for trial trips in which all conditions can be kept favorable; but they are departed from as the steam is throttled and the engines have to work at lower powers than they were designed for. This is particularly the case with war ships which must be provided with a great reserve. It is also likely that the loss from drop and condensation increases with the number of cylinders the steam has to work through, although the initial condensation may not be so great in the first cylinder. Even under the ideal conditions imposed, there is very little gain in quadruple-expansion over a triple-expansion engine at low powers; in practice, there would undoubtedly be considerable loss. Let it be remembered here that the cut-offs cannot be varied much in multiple-expansion engines, and the speed has to be varied by throttling, thus supplying steam to the first cylinder at low pressures for low powers. On the measured-mile trials of the *Yorktown*, the absolute pressure in the boilers was 175 pounds, and the initial pressure of steam in the H.P. cylinder was 164.7 at 16.65 knots, 134.7 at 14.81 knots, 60.7 at 10.62 knots, and 25.2 at 4.4 knots.

The economical speed and endurance cannot be determined accurately for different classes of ships, as no experiments have been made at low powers to fix the value of N_1 and N_2; but the following methods of approximating to their values will be found to give results *substantially correct* at moderate speeds.

$$N_1 = \frac{aD^{\frac{2}{3}}}{C}$$

Compound engines $\begin{cases} a = 2.2 \text{ without jacket,} \\ a = 2.0 \text{ with jacket.} \end{cases}$

Triple-expansion engines $\begin{cases} a = 1.9 \text{ without jacket,} \\ a = 1.7 \text{ with jacket.} \end{cases}$

$C = \frac{L}{3} + 157$, where L = length of ship in feet.

D = mean displacement.

N_2 is equal to the sum of the following quantities:

1. The coal per hour for the main engine auxiliaries. Let H = the maximum power of the main engines.

For auxiliaries driven by simple engines, $b = \frac{H}{29}$.

For auxiliaries driven by compound engines, $b = \frac{H}{35}$.

2. Coal per hour for general auxiliaries and all expenditures for ship's purposes.

$$c = \frac{D}{20} + 132.$$

D = displacement in tons.

3. Coal per hour expended to overcome friction of moving parts of main engines.

For horizontal engines, $d = \frac{410}{R^2} H$.

For vertical engines, $d = \frac{298}{R^2} H$.

H = maximum horse power of main engines.

R = maximum revolutions of screw.

4. Coal per hour required to make up for condensation in cylinders and drop in receivers.

$e = H \times .07$ when the cylinders are not jacketed.

$e = H \times .05$ when the cylinders are jacketed.

5. Coal per hour expended on radiation from boilers and pipes.

Let A = external surface of boilers and pipes in use in square feet.

$$f = \frac{A}{7}.$$

$$N_2 = b + c + d + e + f.$$

These formulæ do not take into consideration the differences of the design of machinery and boilers in different ships, and, of course, they must be regarded as more or less tentative, having been worked up from very meagre data.

Various methods of saving coal have been tried. The *New York* affords a good example of the latest method. Two sets of triple-expansion engines are attached to each shaft with the ob-

ject of throwing out the forward set for ordinary cruising. Some saving will undoubtedly be effected in this way, inasmuch as the engines in use will be more nearly proportioned to the work they have to do, but I think the chief gain will arise from throwing out of use two sets of main engine auxiliaries.

The triple-screw ship is also a good example of the attempt to produce economy at low speeds. The engine in use will be more nearly proportioned to its work, and two sets of main engines and auxiliaries will be thrown out. The drag of two screws may offset the gain in economy to a large extent.

The plan of building a triple-expansion engine with a low-pressure cylinder to be thrown out for cruising and thus working compound, or a quadruple-expansion engine with a low-pressure cylinder to be thrown out, does not possess the advantages of the above two examples, as the efficiency of the main engines will be greatly reduced by reason of lower steam pressures and less expansion within those cylinders in use. All of the main engine auxiliaries will also have to be run.

In the matter of design, it strikes me that altogether too much attention has been paid to jacketing the barrels of the cylinders, especially those of large diameter in proportion to the stroke.

The high-pressure cylinder of a triple or quadruple-expansion engine should be well jacketed all over, and other cylinders should be jacketed only in the heads. It is the clearance surfaces that cause the greater part of steam condensation in the cylinders. Improvement here would take the form of a coating of non-conducting material on the cylinder heads and pistons. The L.P. piston of the *San Francisco*, which is 94 inches in diameter, has to move 23½ inches, or two-thirds of the stroke, to uncover a cylindrical surface equal to the area of one cylinder head. In point of time one square foot of clearance surface is exposed to the alternate action of live steam and exhaust 2.3 times as long as one square foot of cylindrical surface, and since, in the above case, the former is 145 square feet, while the latter is 52 square feet, the clearance is 6.4 times as effective in producing condensation as the cylinder proper. It would have been far better to dry the steam before admission to the cylinders by

means of a nest of steam pipes in the receivers than to waste it in the cylindrical jackets.

The enormous clearances that inevitably accompany piston valves on L.P. cylinders cause great loss, and point to some improved design of the slide valve as the next step. The wear of piston-valve chambers and rings invariably increases the steam leak through the engine, and again emphasizes the superiority of the slide valve. If some method of balancing the latter and taking up the pressure required to overcome the inertia of the reciprocating parts of the valve gear were devised, there could be no question of its supplanting the piston valves on all of the cylinders.

As already stated, the auxiliaries and losses have not received the attention they deserve. Very little, if anything, is gained in making the auxiliary engines compound, unless they are designed to work at their maximum powers as compound engines while the main engines are running at low speeds. If a live steam pipe be led to the L.P. cylinder, and the H.P. cylinder be provided with an exhaust to the condenser, so that each auxiliary may have a large reserve of power for high speeds of the main engine, this plan may be made very efficient.

Probably a better plan would be to make the main engine auxiliaries simple, with steam cylinders large enough to run at low steam pressures, and then incorporate them into the main engines by taking steam from the I.P. receiver and exhausting into the L.P. receiver. A live steam pipe and separate exhaust to the condenser could be provided for increase of power. The above would amount to an increase in the size of the intermediate cylinder, and to making the auxiliaries part of a triple-expansion system, precisely as if they were coupled to the shaft of the main engines. If live steam pipes were run through the receivers to correct liquefaction, I see no reason why the engine-room auxiliaries could not be made as economical as the main engines. The only objection to this plan lies in the variation of pressure in the two receivers, but this variation is not great at cruising speeds. The *Yorktown*, for instance, had a difference of 13 pounds between the intermediate and low pressure receivers when running at 10 knots.

Cross connections have been fitted in some cases, so that the pumps of either engine room can be used for both at very low speeds.

In conclusion, I will say that any careful observations upon the various coal expenditures on board ship would be of real benefit to the service. It is difficult to obtain a large ship in commission for experimental purposes; besides there is considerable expense attendant upon experiments that would be of any use. Many opportunities occur during a cruise, however, when important data can be taken without extra expense to the Government, if officers can spare the time for such work. Such data forwarded to this JOURNAL for publication would add greatly to its value, and also serve to clear up the entire question of coal expense.

DISCUSSION.

Chief Engineer Isherwood, U. S. Navy.—The problem proposed by Mr. Hollis is so difficult under the stated conditions, that its satisfactory solution, without additional specially determined experimental data, seems impossible, owing to its many variables, all of which vary widely, and most of which are unknown quantities. The best that can be done is a roughly approximate determination effected more by what may be called the instinct of practical experience than by any methods of exact calculation. Indeed, with only the usually known data, no reliable inferences can be made by calculation alone from the performance of steam machinery and vessels at one speed, to what would be their performance at other considerably different speeds.

The problem is simply to ascertain for the case of a given steamer the speed at which the vessel must be propelled in order to pass over the greatest number of miles, irrespective of time, with the coal carried in the bunker. This is a very important problem for naval vessels, as the time occupied by them in going from one port to another is generally of small consequence in time of peace, while the coal consumed in making the passage

is very important, not only from its money cost, but from the fact that the vessel's steaming distance is limited by the quantity of coal carried in the bunker.

Now, in a general way, and for speeds at which the vessel does not "squat" or alter trim, the power to be developed by the engine varies as the cube of the speed of the vessel, or, in other words, eight times the power is required to double the speed; and, as the unit of power for the same engine may be considered, in a general way, to be obtained by the consumption of the same weight of coal, eight times the weight of coal is required to be consumed per hour to double the vessel's speed. If, then, the speed of any vessel be halved, the consumption of coal per unit of time and the power developed will be decreased to one-eighth, but as the distance gone will be only one-half, the coal expended per unit of distance will be one-quarter of what it was; consequently, a given weight of coal will drive the vessel four times as far at half speed. Hence, the distances the vessel will be driven by the consumption of a given quantity of coal at different rates will be as the inverse squares of the coal consumed per unit of time. Consequently, according to the above purely theoretical assumptions, the lower the speed of the vessel the farther will the consumption of a given weight carry it, and in the ratio just shown.

The theoretical assumptions are that the resistancee of the vessel varies as the cube of its speed; that the resistance is always the same at the same speed; that the power developed by the engine is as the cube of the speed of the vessel; that either the whole of this power, or a constant fraction thereof, is applied propulsively; and that the consumption of coal is in the direct ratio of the power developed. But as these assumptions are so far from corresponding to the facts of experience that any inferences drawn from them will be widely erroneous, an attempt must be made to apply to them the corrections of practice. This Mr. Hollis has done in a masterly manner, and if his method does not precisely solve the problem, it is because the problem is insoluble under the conditions of his statement. He has done all that is possible with them; to do more requires more data.

As coal is consumed on board a steamer for many purposes other than the propulsion of the vessel, such as lighting, warming, ventilating, steering, bilge pumping, distillation of water, etc., and, as the coal for all these purposes is consumed when the motive engine is in operation, in the same boiler and during the same time, the respective quantities consumed for propelling the vessel and for the other enumerated purposes cannot be separated, and consequently cannot be known. The best that can be done is to make a practical estimate of the coal required for the latter, which coal will be nearly constant per day, let the vessel's speed be what it may, and then making an estimate of the number of days the vessel will require to make the intended passage at any assumed speed, deduct from the weight of coal in the bunker the weight of coal to be expended for other than the propulsion of the vessel, considering only the remainder as the weight of coal carried in the bunker for propelling use. Evidently this remainder will be different for every assumed speed of the vessel; the less the speed the less will be the remainder of coal for propelling the vessel. The difference thus caused is important in the solution of the problem. The less the weight of coal carried in the bunker, and the greater the weight of coal consumed for other purposes than propelling the vessel, the more widely will the remainder coal vary for different speeds of vessel.

If the weight of coal consumed per day for the foregoing-enumerated purposes other than the propulsion of the vessel was any constant percentage of the weight required for propelling, there would be no necessity for its separate determination and subtraction; but as it is, on the contrary, nearly a fixed quantity under all circumstances, it becomes more and more important as the weight of coal expended for propelling the vessel becomes less and less; hence its separate determination and subtraction from the total coal carried is imperative as a first step.

The difficulty is to ascertain the weight of coal expended for other than propelling purposes. It can be approximated for any given vessel by using one boiler when in port, or when the motive engine is not in operation, to furnish the steam required for

lighting, heating, ventilating, water distilling, etc., and carefully weighing the coal consumed per day. The mean of this weight for several days' use will be as close an approximation as can be made with the present method of boiler arrangement, which furnishes steam indiscriminately for the motive engine and for other uses.

A much superior method in every particular would be the following: Place in the vessel a separate boiler of sufficient steam-generating capacity to furnish steam for all purposes except propelling the vessel, and make all the steam connections to all the auxiliary engines, and to the heating and distilling apparatus from this boiler alone, which is to be of a type and proportions that will enable it to carry a steam pressure of, say, 25 to 30 pounds per square inch greater than the boiler supplying steam to the motive engine. The auxiliary engines, supplied with steam from this higher-pressure boiler, are to exhaust into the lower-pressure boiler supplying the motive engine. By this system the only heat expended for the auxiliary engines will be what is theoretically necessary for conversion into the power developed by them. There will be no rejected heat lost. The whole of the heat in the steam drawn from the higher-pressure boiler to the auxiliary engines will be returned into the lower-pressure boiler to be used in the motive engine, less only the thermodynamic quantity expended in the production of the power of the auxiliaries. This quantity will be so small as to be practically negligible, and the entire power developed by the auxiliaries will be obtained for sensibly nothing. Such a method at once eliminates the coal used for the auxiliary engines from the problem, besides effecting an important economy in the production of the auxiliary power. In the auxiliaries the steam would be worked without expansion, and those engines would be of the simplest type, as nothing would be gained either by using the steam expansively or by employing complex types of engine.

In the old navy, where the lighting was done by a few sperm candles and oil lamps instead of by dynamos, where there was no artificial ventilation except what was produced by wind-sails,

where keeping warm was effected by pea coats and not by steam heaters, where bilge pumping and steering were done by manual labor and not by steam engines, and whose potable water was carried in tanks instead of being supplied by the distillation of sea water, this question of the coal now consumed for these operations did not enter into the problem. In the old Navy there were no auxiliary engines to the motive engine. The motive engine worked its own air pump, circulating pump, bilge and feed pumps, so that the influence of separate auxiliaries for these purposes did not complicate the problem. With modern vessels what, for brevity, may be called the pumping, is done by auxiliaries to the motive engine, which alone is applied to the propulsion of the vessel. As the pumping required is, roundly, in proportion to the motive power exerted, the coal required for the auxiliaries should be directly proportional to what is required for the motive engine. Of course, the horse power is obtained for a much greater weight of coal per hour in the case of the pumping auxiliary engines than in the case of the motive engine, but this does not affect the proportionality of the consumption, and as long as the *proportion* remains constant, the *absolute quantity* does not affect the problem. The "blowing" engines, or engines driving the blowers which supply the furnaces with air, can be omitted in this investigation, as at any economical speed of vessel the power will be easily obtained by the natural draft of the boiler.

The result is thus reached that the fixed or constant expenditure of coal per day, additional to that required for the motive engine and its pumping auxiliaries, may, with reasonable approximation, be obtained from the consumption when the vessel is at anchor in port, and that the consumption for the motive engine and its auxiliaries will be in the ratio of the power developed by the motive engine alone. Hence, taking the weight of coal remaining in the bunkers after subtraction of the weight required for the fixed expenditure appertaining wholly to the vessel, and not at all to the motive engine, as the true weight of coal carried for the propulsion of the vessel, the calculation of the greatest distance to which this remainder can propel the vessel, irrespect-

ive of speed, could be made with sufficient closeness, provided the resistance of the vessel at any given speed remained always the same, and at different speeds remained in a constant known ratio to the speed, and provided that the cost of the horse power in weight of coal consumed per hour also remained always the same. But none of these assumptions is true.

The resistance of the vessel varies with the more or less cleanness of its immersed external surfaces, with its trim and draught of water, with the direction and force of the wind, with the state of the sea, and, after the speed within which there is no "squatting" is passed, the resistance varies from the theoretical law of the square of the speed, and, in a constantly-varying degree, the ratio becoming higher and higher as the speed becomes greater and greater. And this ratio is not only unknown, but it varies with the size of the vessel and form of the immersed solid.

For long intervals of time, such as required to pass between distant ports, involving the maximum steaming endurance of the vessel, the cleanness of bottom, state of wind and sea, etc., may be considered to about average the same.

As all economical speeds are within the limit of no squatting, the resistance of the vessel under the same conditions of surface cleanness, wind, sea, etc., may be taken without error to be in the ratio of the square of the speed, making the power for sameness of these conditions to be in the ratio of the cube of the speed. Of course there must be known the power required to propel the vessel at some speed less than the squatting limit, and the consumption of coal therefor, motive engine and auxiliaries included.

The slip of the screw remains constant for the same conditions of vessel for all speeds below that of the squatting limit, but varies for the same speed if these conditions be not constant; and the slip of the screw affects, *pro tanto*, the economy of the power—the less the slip, the greater the economy. Also, the power consumed in overcoming the surface resistance of the helicoidal surface of the same screw to the water is greater with the greater slip for equal speed of vessel. Thus, the variability of the slip of the screw complicates the problem, and may, in extreme cases, to a considerable extent.

The weight of steam produced by a given weight of the same coal consumed in the same boilers, supposing them to be of the type and proportions in ordinary use for sea-going vessels, will vary with the rate of combustion, increasing as the weight decreases, down to the limit of about half a dozen pounds of coal per hour per square foot of grate surface. The economic vaporization of the boilers is affected, too, by the pressure of the steam carried in them. Other things equal, the higher the pressure, the less in some unknown ratio will be the weight of water vaporized per pound of coal. And herein will be found other disturbing factors of still greater effect.

On the other hand, the economy of the power in weight of steam consumed lessens as the amount of the power developed by the engine lessens. The consumption of steam for the motive engine, other things equal, is proportional to the *total* (not the net or the indicated) power developed by it—that is, to the power calculated for the mean cylinder pressure measured down to zero. The power, however, which propels the vessel is only a fraction, and a widely varying one, too, of this total power, being only what remains of it after all the powers employed in overcoming the frictions of all kinds, including screw surface, as well as the power required to overcome the back pressure against the piston, are deducted. If the pumping is done by the motive engine, the power to do it must also be deducted, but not otherwise. Now, the back pressure and the friction pressure of the same unloaded engine may be taken to be the same under all circumstances, but the friction due to the load, and the resistance of the screw surface to the water, may be taken as proportional to the power which remains of the total power after deduction of the powers required to overcome the back pressure and the friction of the unloaded engine. Hence, the disturbing factor, as regards pressure, is the varying ratio of the sum of the back pressure against the piston, and of the pressure required to work the unloaded engine, both of which should be constant, let the total power developed be what it may, while the remaining portion of the total power—namely, the portion which overcomes the resistance of the screw surface to the water and the friction

of the load—besides being further expended in the slip of the screw and in the propulsion of the vessel, varies in the ratio of the cube of the speed of the vessel, other things equal. As the portion of the total mean cylinder pressure which is applied to the propulsion of the vessel becomes smaller and smaller in consequence of reductions in the speed of the vessel, the cost, in function of cylinder pressure, of that portion in coal consumed per hour becomes greater and greater; because, the total expenditure of coal being proportional to the total power developed, the smaller the fraction of the total power utilized propulsively, the greater will be the cost of that fraction per unit, as the entire coal expenditure is chargeable to it.

Thus far, the greater fuel cost of the effective horse power, as it becomes lessened by decrease of the speed of the vessel, has been considered as affected by only the lessened fraction which the effective is of the total pressure on the piston, as the speed decreases. When carried far, the increase in the cost of the effective horse power under these conditions, may become enormous. For example, suppose that at a certain speed of a given vessel, the total mean cylinder pressure is thirty pounds per square inch, and that the fixed sum of the back pressure and pressure required to work the unloaded engine is five pounds per square inch, the net horse power would be represented by (30—5=) 25, and the expenditure of coal by 30. Now, let the speed be so far reduced that the total mean cylinder pressure falls to ten pounds per square inch, the sum of the back pressure and pressure required to work the unloaded engine being, as before, five pounds per square inch, then the net horse power would be represented by (10—5=) 5, and the expenditure of coal by 10. At the high speed, the cost in fuel of the net horse power would be represented by ($\frac{30}{25}$=) 1.2, and, at the low speed, this cost would be represented by ($\frac{10}{5}$=) 2.0 or two-thirds more.

But the employment of lower speeds of vessel entails not only an increased cost of the effective power in function of pressure, as just shown, but a still greater cost owing to the corresponding reduction in the reciprocating speed of the piston.

The less this speed and the less the total mean pressure in the cylinder the greater will be the proportion of cylinder condensation, and in a very marked degree for engines using saturated steam without efficient steam jacketing. Combining the two causes of lessened economy, the cost in fuel of the net horse power may easily rise in practical cases of much reduction of speed to double the cost at the higher speed. If the cylinder be efficiently steam jacketed, or the steam be sufficiently superheated, the loss due to the higher ratio of cylinder condensation at reduced speeds can be avoided.

In engines which, though designed for use at high speeds of vessel, are largely used at low speeds, it is of the utmost importance that they realize maximum vacuums, that their cylinders be thoroughly steam jacketed, and that superheated steam be employed. The back pressure against the pistons and the liquefaction of steam in the cylinders should be reduced to the minimum; and the value of these reductions becomes greater and greater as the mean total pressure in the cylinders becomes smaller and smaller.

In steam engineering one factor cannot be changed without changing all the others, and the complexity thus introduced makes accurate calculation in such cases as that of the increased fuel cost of the power at decreased speeds of vessel almost an impossibility.

Experiment then becomes the only guide, as the value of every quantity depends wholly on the limitations. If, in any given vessel, the fuel cost of the horse power at different speeds was experimentally known, and also the quanity of fuel used for the purposes of the vessel other than its propulsion, calculation could do its work in determining the most economical speed; that is to say, the speed at which, with a given weight of fuel to be expended, the vessel could pass over the greatest distance. Wanting this data, however, the writer would advise the following method, which supposes the vessel to have already done considerable cruising, and that an accurate log had been kept of the performance:

Take from the log of the performance of any given vessel all

the data for speeds between, say, $6\frac{1}{2}$ and $7\frac{1}{2}$ knots, and consider the mean as the data for the speed of 7 knots. In like manner take all the data for speeds between $7\frac{1}{2}$ and $8\frac{1}{2}$ knots, and consider the mean as the data for the speed of 8 knots, and so on, up to the highest speed, advancing by increases of 1 knot. There would thus be obtained the consumption of coal in pounds per hour due to all uses corresponding to these speeds, which weight of coal, divided into the total number of pounds carried in the bunkers, gives the number of hours steaming at the speed taken, and this number, multiplied by the speed, gives the distance the vessel will pass over with the total weight of coal carried in the bunkers. This is the most direct, sure and simple method of ascertaining the most economical speed. If the speeds of the vessel be laid off by scale, as abscissæ on a straight line, and if, on ordinates erected from these abscissæ at right angles to the line, there be laid off by scale the corresponding consumption of coal, a graphic curve passed among the free ends of the ordinates so as to leave as many on the one side as on the other, would serve as a formula of interpolation from which the coal consumption for any speed between the experimental limits could be obtained. The other data, such as the number of revolutions made per minute by the screw, the various steam pressures, vacuum, temperatures, etc., etc., should be similarly obtained and recorded. If the cruising thus digested has been considerable, the differences due to wind and wave, cleanness of the bottom of the vessel, variations of draught of water, etc., will go far to offset each other, and will disappear in the final averages. The greater the amount of cruising, the more accurate will these latter be. This is what should be done and at once for all cruising naval vessels, and the results furnished to their commanding officers and chief engineers. The task is long and laborious. A sufficient number of assistant engineers should be detailed for this duty, the result of which is really of very great value to the Navy. Mr. Hollis has done excellent work in bringing this subject prominently forward, and it should not be allowed to lapse into the abyss of wise suggestions neglected and forgotten.

Dr. Charles E. Emery, Member.—The paper of Mr. Hollis treats of a very interesting and valuable subject of investigation. It may be stated in general terms that when the "ship's auxiliaries," as he terms them, are proportioned, as they must be, for maximum power, the expenditure therefor must necessarily form a larger proportion of the whole expenditure for lower powers. The only remedy seems to be a number of units of each kind, whereby less auxiliary cylinder capacity can be employed at lower speeds, or, what is much the same thing in effect, the auxiliaries may be operated in series with one or more of the main cylinders at slow speeds, thus using a difference of steam pressure at such times, and permitting the use of the entire range of pressure for the higher speeds, the cylinders being of reduced size proportioned therefor. The complication due to multiplication of units cannot be permitted to any great extent, and the complication of connections to use different pressures, though less objectionable, is nevertheless subject to criticism. However, by the continuous exercise of good judgment, economy can be obtained by the use of either or both of these methods. It is hoped that the progressive officers of the service will follow this matter up until a complete solution is reached. The final outcome may be that if the auxiliaries necessarily use such a large proportion of the coal supply, a simplification of the construction of the main engines may be permitted. It will be recollected that some of Mr. Isherwood's experiments with the Herreshoff boats showed economies for compound engines closely approximating those afterward obtained with triple compound engines. A recent paper by Messrs. Green & Rockwood* states the results of experiments with a triple compound land engine in comparison with results from the same engine run as a compound engine with the intermediate cylinder out of circuit. There were two intermediate re-heating chambers used in the first case which were joined in series in the second. Perhaps the proportions were not the best possible for a triple compound engine. The feature, however, is that such remarkable

* "Two Cylinder *versus* Multi-Cylinder Engines," by S. M. Green and G. I. Rockwood, Vol. XIII, Trans. Am. Soc. M. E., page 647.

economy could be obtained with a compound engine of any type. The results were: Compound engine trials, 13.41 pounds and 13.11 pounds; triple compound engine trials, 13.01 pounds and 13.25 pounds of water per indicated horse power per hour, including the steam used in the jackets and heaters. The average steam pressure was 142 pounds; the indicated horse power 180 to 190. Such results as these would indicate the possibility of dispensing with the weight of the intermediate cylinder, except where from the size of the engine three cranks are necessary and where a better distribution of the load can be made triple compound than with three-cylinder compound. If, however, the triple compound engine be the more economical under the conditions of marine practice where the complication of independent cut-offs is undesirable, the addition of a constant charge due to auxiliaries tends to reduce the percentage between the two types of engine, and as a final result the compound engine may from reduced radiating surface be the more economical, and from reduced weight be otherwise more desirable for all purposes except maximum speed, and lose so little under such conditions as to warrant the change. This "heresy" is not only warranted by the experiments above referred to, but by the conditions imposed by the auxiliary apparatus. Notice, however, that the high pressure used with the triple compound engine must be maintained for the compound. It may appear that, for a pressure of 160 pounds or under, the distribution of temperatures may be so arranged in two cylinders that with proper reheating the cylinder condensation may be so reduced that substantially similar results will, as in case above cited, be obtained as by distributing the temperature through three cylinders.

I have obtained recently some very curious results on a commercial basis in regard to steam engines of different types for use on land. Commercially, the interest on the cost of steam machinery is as much of a burden to the owner as the coal, and I have found, by a careful summation of the various items of cost for different types of engines now in use on land, particularly those adapted for generating electric current, that, granting all the saving in coal claimed for the more economical engines, the

other items of cost, such as wages, insurance and taxes, add a practically constant quantity, which greatly reduces the relative percentages of saving for the higher priced engines, and, when the interest is considered, the saving is neutralized, and compound engines giving fairly good results show as well in the final comparison as triple compound engines of elaborate construction, necessarily furnished at a higher cost. This paper will appear in about a month in the "Transactions of the American Institute of Electrical Engineers," and, I trust, will be studied with a great deal of interest by my former associates in the Service, and others who have had the opportunity of taking up their work under better auspices. Cost can hardly be considered in a naval vessel, where the highest efficiency secures so many advantages, but the auxiliary costs may, as in the case mentioned, be sufficient, with the new facts available, to warrant a change in the type of machinery when all the questions of performance are weighed against each other.

Asa M. Mattice, Esq., Member.—The paper under discussion points out in a very clear manner some rather startling truths. Of course we all knew that, as the power of the main engines decreases, the relative power of the auxiliaries increases, but nobody who has not gathered a mass of valuable data and systematized it, as Mr. Hollis has done so admirably, could have believed that figures derived from actual practice would show such remarkable results.

Mr. Hollis' point on the relative efficiencies of head and barrel jackets is well taken. On large short-stroke cylinders, if any jacket is to be omitted, it certainly should be the barrel jacket. Not only are the head jackets more economical, for the reasons pointed out by Mr. Hollis, but because the omission of a jacketed liner decreases the clearance, and because with large diameters and short strokes the head jackets can be put in on less weight than barrel jackets. When possible, moreover, the steam passages should also be jacketed, especially if they are long. This is the part of the cylinder with which the alternately hot and cold steam comes most intimately into contact. The last report

of the English "Research Committee" cites a good example of saving by an engine being jacketed all over, valve chests and all, the engines in the case showing a much larger gain from jacketing than when the cylinders and heads alone are jacketed.

As to reheating in the receivers, which Mr. Hollis commends, I cannot understand why this system had not long ago been adopted in marine practice. All high-grade stationary engines have either tubular reheaters or very large steam-jacketed receivers, without which they could not attain the high duties that they do. E. D. Leavitt has used them for at least thirteen years, and I believe that he would about as soon think of omitting the pistons or crank shaft as the reheaters. Although the tubular reheater is the best, yet there is a considerable gain by the ordinary jacketed receiver, usually made of boiler plate. These do not weigh much, and in marine work could usually be suspended from the deck beams.

The question of non-conducting surfaces for pistons and cylinder heads is a long-agitated subject, and I have no doubt that something practical will be done in this line before many years.

As to the relative merits of piston and slide valves, I agree with Mr. Hollis that the slide valve can be made the more economical instrument. Its best form is not to be sought, however, in the old type of valve, where all the functions of inlet and exhaust are performed by the one clumsy, bulky and heavy casting. The most economical slide valve, in fact the most economical valve of any kind, is the gridiron, with four valves to a cylinder, two for inlet and two for exhaust. The four valves do not begin to weigh as much as one old-style valve giving the same port opening, and have the advantage of wearing tight while the other wears leaky. The unequal wear of the ordinary slide valve is due to the varying unbalanced pressures at various parts of its bearing surface, while, with the gridiron valve the unbalanced pressure is equally distributed all over and the wearing surface is very large. Gridiron valves are in use which are as tight to-day as when they were first scraped-in sixteen years ago, without a particle of work having been done on them. Then, the motion of the gridiron valve can be made very small, especially if a cam-

valve gear be used. Thus, for 1½ inches width of port opening, a total valve travel of 2⅛ inches is sufficient, which is rather smaller than the travel required with either an ordinary slide or a piston valve. Besides, the gridiron, if properly made, will give less total cylinder clearance than any other valve made. With a cam gear the cut-off can be varied from zero to 95 per cent. of the stroke, with full valve opening from 5 per cent. to 90 per cent. of the stroke.

When I speak of a cam gear, I mean a scientifically constructed cam gear. The idea which many people seem to have of a cam is that it is a thing to hit something else and knock it out of the way; and very many cams seem to have been constructed on that principle. A properly designed cam will start and stop a valve as easily as if the motion were imparted by an eccentric, at the same time giving a very quick motion and a sharp cut-off. As to the durability of cam gears, I may cite a case where seventeen engines with gridiron valves and Leavitt cam gear have been working night and day, 144 hours a week, for over ten years, at from 98 to 100 revolutions, and the cams and gear look as if they were good for about a century more.

There are a few additions that I would like to see made to the data obtained on full-speed trials as well as at lower speeds, in order that the performances may be more fully analyzed. Possibly this may have been done, but I have not seen it in any of the published reports. What I refer to more particularly is the amount of feed water, the dryness of the steam, and the temperature of the gases in the uptake. Of course the measurement of feed water by tanks is entirely out of the question, except in a ship detailed for experiment. But water meters are now made which, if carefully calibrated and properly fitted, will give the volume of feed water to within two per cent., which is much better than knowing nothing about it. The meter can be part of the regular engine-room equipment and its reading a regular entry in the log at all times. It might be fitted with a bye-pass to be used for feeding direct in case of possible derangement. The principal obstacle to its successful use is the

pressure of air, and it is therefore necessary that provision be made to exclude all air from the feed pipe, or to remove it before reaching the meter. The dryness of the steam should be measured at two places—at the boiler and at the engine. At low speeds the throttling calorimeter would in nearly all cases give sufficiently accurate results, while at full speed, when the steam is probably very wet, a separate calorimeter would be needed. However, as both of these instruments together take up very little room, it would be well to fit them both on all ships. The uptake temperatures could be taken very easily by means of pyrometers. It would require two additional observers to obtain the data here suggested on full-power trials, but the value of the information in future designs would be almost priceless. Valuable information could also be obtained by collecting samples of the uptake gases for analysis. A great deal would be learned in this way.

Now, as to the economy of the engines at cruising speed. As Mr. Hollis tells us, the best plan so far put forth is to have two complete triple-expansion engines for each screw, disconnecting one of them when cruising. The disadvantage of this is weight and space. Cutting out the low-pressure cylinder helps out the economy to some extent, but not as much as the former method. I have studied over this problem for some time, and I think that the solution suggested below is here given for the first time. It is almost self evident that the desired result can only be obtained by the use of a smaller engine at cruising speed than at high speed, but to make this practical it must be done with little, if any, increase of weight and space.

My plan consists in adding a small cylinder to be used only while cruising. This cylinder would then be the H.P. cylinder; the high speed H.P. cylinder would become the cruising I.P. cylinder; the high speed I.P. cylinder would become the cruising L.P. cylinder; while the high speed L.P. cylinder would be cut out by a stop valve and by disconnecting the connecting rod. Take the case of the *Yorktown*, for instance. From the tables given by Mr. Hollis in his lectures before the Naval War College, it appears that the engines were most economical at 14.81

knots and 140.6 revolutions, where the coal per I.H.P. per hour was 1.9 pounds, while at 8 knots and 72.3 revolutions the coal was 2.6 pounds. The indicated horse power of the main engines was 2,217 at the higher speed, and only 310 at the lower. The L.P. piston displacement at 8 knots was 31.8 cubic feet per I.H.P. per minute, which is altogether too high for economy, while at 14.81 knots it was 8.64 cubic feet, which is not far from that of the best performances. By plotting the various speeds and coal per I.H.P. given by Mr. Hollis (see p. 671 of this JOURNAL for November, 1892) it will easily be seen that there must be a speed, probably between 13 and 14 knots, where the coal was slightly less than 1.9 pounds per I.H.P., perhaps getting as low as 1.8, and with a L.P. piston displacement per I.H.P. of not far from 10 cubic feet per minute.

The *Yorktown's* cylinders are $\frac{22'' \times 31'' \times 50''}{30''}$, and the ratios are $\frac{IP}{HP} = 2$, $\frac{LP}{IP} = 2.61$, $\frac{LP}{HP} = 5.22$.

Now, let us add a supplementary cylinder of, say, 14 inches diameter, to remain idle at full speed, but to be connected up for cruising, and the 50-inch cylinder disconnected. We now have an engine with cylinders $\frac{14'' \times 22'' \times 31''}{30''}$ with ratios

$$\frac{IP}{HP} = 2.5, \quad \frac{LP}{IP} = 2, \quad \frac{LP}{HP} = 5,$$

and at eight knots the L.P piston displacement per I.H.P per minute would be only 12.2, instead of 31.8, cubic feet, and as the best authentically recorded marine performance, that of the *Iona*, was made with 10.84 cubic feet L.P. piston displacement per minute, we would not be far out of the way. This supplementary cylinder should be thoroughly jacketed all over and preferably fitted with four valves, with a large range of cut-off. Next to a gridiron valve, I would prefer a valve of the Corliss type. To be sure, Corliss valves and gear were tried at sea with not very good results many years ago, but great improvements have been made in the gear since then, so that the old objections cannot be applied to the present gears.

At full power, the main engines of the *Yorktown* develop 3,416 I.H.P., which, if the work is evenly divided, gives about 570 I.H.P. to each cylinder. At 8 knots, the I.H.P. of each cylinder would be about 52, so that the working parts of the supplementary engine would only have to be less than a tenth as strong as for the remainder of the engine; or, if we wished to make these parts strong enough for 10-knot cruising, or say something less than 100 I.H.P. per cylinder, the working parts would not have to be over one-fifth as strong as the others. It will be seen, therefore, that the supplementary engine could be built with very little weight. In a great many cases, especially with vertical engines where the H.P. valves are abaft the cylinder, the supplementary cylinder could be so placed that it could work on an overhung crank on the forward end of the shaft, so that no extra shaft or bearings would be necessary. The crank shaft, being proportioned for full power, would always be large enough for this overhung crank, which is to be used only at low power.

In all cases an engine built on this system would give a smaller cruising engine than if there were two engines of equal size to each screw, thus giving greater economy at low speeds, besides saving weight and space; also having the advantage that it can be used where the other is impossible. Take the case of the *New York*, for instance. For each screw she has two engines $\frac{32'' \times 46'' \times 70''}{42''}$, with the following ratios, $\frac{IP}{HP} = 2.07$, $\frac{LP}{IP} = 2.31$, $\frac{LP}{HP} = 4.78$. If we put this into one engine we would have cylinders $\frac{45'' \times 65'' \times 99''}{42''}$. Now, let us add to these a supplementary cylinder which shall be 40 per cent. of the area of the 45-inch cylinder, or say 28-inch diameter.

The annexed diagrams will show the arrangement of the engines for full power and for cruising, the arrows showing the course of the steam.

I do not know the size of the *New York*, or what speed she is expected to make with the designed 16,000 I.H.P., but it is pretty safe to say that the power required for ordinary cruising would

Arrangement for Full Power.

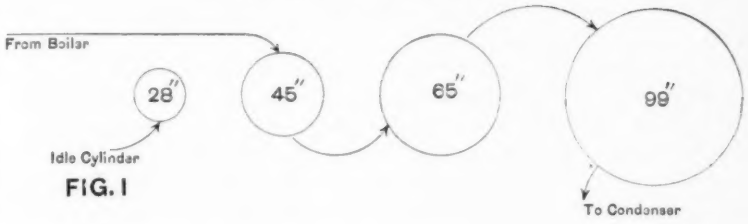


FIG. 1

Arrangement for Cruising.

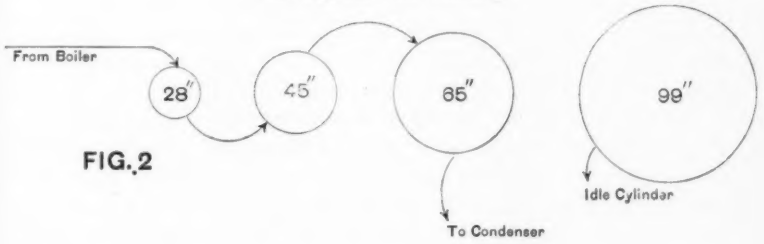


FIG. 2

Arrangement for Full Power.

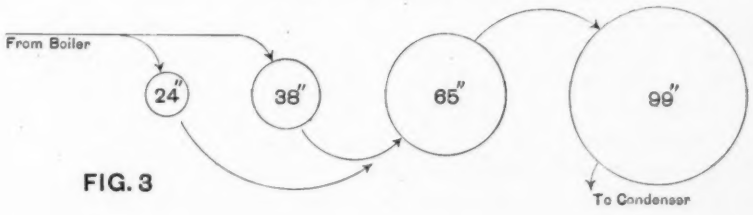


FIG. 3

Arrangement for Cruising.

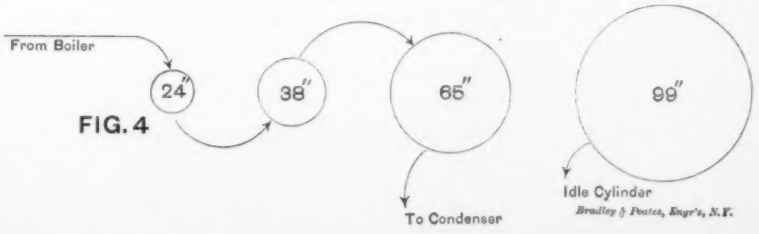


FIG. 4

Bradley & Peates, Engr's, N.Y.

not exceed one-eighth of this at the outside, so that the working parts connected with the 28-inch cylinder would only have to be one-eighth as strong as those of the other cylinders as here shown, or one-fourth as strong as the working parts of the two separate engines, now built for the vessel. As the 28 inch engine would not be called upon to develop much more than 300 I.H.P., it will readily be seen that the weight of that part of the machinery would not be very great. This power could readily be put on an overhung crank, as before suggested.

Moreover, whenever space would permit, I would supply a single vertical bucket-air-pump of suitable size for cruising speed, worked by a beam off the supplementary engine, with feed pumps and bilge pumps attached. At cruising speed there would be no difficulty in working the air pumps in this way; in fact, they are still worked by the main engine on many of the Atlantic "fliers." An ejector could be fitted to maintain the vacuum when starting the engines, &c. By working small air pumps in this way for cruising, we would not only save the large amount of friction which is necessary when we work the large independent air pumps for low powers, but we would derive the power for working them from a much more economical engine. For circulating pumps it would, of course, be necessary to use independent centrifugal pumps, but for each main engine I would have two of these, one large and one small; both to be used at full power and the smaller one for cruising. This might not be possible on the smaller vessels, but could nearly always be arranged on the larger ones. In the case above cited, of a vessel the size of the *New York*, if not desirable to use a cylinder as large as 99-inch, we could use two 70-inch L.P. cylinders; both being thrown out when cruising, but even then, with our supplementary cruising cylinders and air pumps, but with one large condenser in place of two smaller, and with one independent air pump and engine, we would save a considerable amount both in weight and space.

In some cases where fore-and-aft space would not permit of the supplementary cylinder being placed forward of the others, it could be placed at one side, and drive one of the regular

crossheads by means of a beam the links of which would be thrown off when not in use. As the power developed by the cylinder would always be small, this could easily be done, and the air pump worked from the same beam at a shorter radius. In other cases it might be preferable to run a tail rod up from one of the main cylinders and key the piston rod of the supplementary cylinder to it when in use, this cylinder being placed above the protective deck if necessary, as it would not be used in action.

A modification of the system above described, which has some advantages, is to make two H.P. cylinders of different sizes, both to be used at full power. In the case before cited, for a vessel like the *New York*, the 28-inch and 45-inch cylinders could be replaced by a 24-inch and 38-inch, arranged as follows:

The 24-inch and 38-inch cylinders are together equal to a 45-inch, and the cylinder ratios for full power are the same as before. The working parts of these engines would be smaller than those of the 65-inch and 99-inch cylinders, as the two together would only develop the power of one of the others. By throwing out the low pressure cylinder and connecting the others, as shown in Fig. 4, we would have an economical cruising engine.

By this method a better ratio of cylinders is practicable than by the arrangements first mentioned. The ratios in Fig. 4 are:

$$\frac{IP}{HP} = 2.5, \quad \frac{LP}{IP} = 2.93, \quad \frac{LP}{HP} = 7.32.$$

At two-thirds power it might be found desirable to throw out the 24-inch cylinder, and work triple expansion with the three other cylinders—that is to say, with the 38-inch, 65-inch and 99-inch. Then, again, it might be found profitable at some lower speed, but not as low as ordinary cruising speed, to lengthen the cut-off of the 24-inch cylinder and work quadruple expansion with all four cylinders.

In the various examples, I have simply approximated to the size of the small cylinder. The exact size would, of course, have to be worked out for each case after deciding upon a cruising speed for which the engine should be made the most economical. In a marine engine, the ratio of cylinders is more a matter

of compromise between equal crank effort, equal range of temperature, and minimum "drop" than is the case with stationary engines, on account of the large range of working conditions. When, however, the supplementary engine is to be used, the speed and power are so small that equal crank effort may be neglected, and the cylinder proportioned best to meet the requirements of "drop" and temperature.

The principal obstacle in the way of applying the supplementary cylinder system to our war vessels is the slight increase of weight. This, of course, does not apply to cases where there is sufficient weight allowed to put in two complete triple-expansion engines for each screw, for in such cases we would save weight by using the system here proposed. In any case if by adding fifty tons to the weight of machinery we can save 100 tons of coal for every filling of the bunkers, common sense would dictate putting in the fifty tons extra machinery by all means. But if the same system is now carried out at the Navy Department as when I was on duty at the Bureau of Steam Engineering, this would be impossible. The Bureau was given a certain weight and space, and directed to put in machinery of a certain power or to drive the vessel at a given speed. There was no possibility of a "give-and-take" between machinery and coal, for the coal supply was determined by other considerations. An attempt was made at one time to readjust this on a common-sense basis, and give to the Bureau the discretion of dividing the total weight allowed for machinery and coal to the best advantage, but was not carried out. When the Navy is run in the best interest of the Government, things will be different, and the Bureau will be free to build even better machinery than the admirable engines that it has already turned out.

The 14-inch by 30-inch cylinders proposed for a vessel of the size of the *Yorktown*, with all working parts and framing, could not weigh, even with a liberal use of materials, over 20 tons for both engines. The saving would be, at least, the difference between 2.6 pounds of coal per I.H.P. per hour and 1.9 pounds, as given in Mr. Hollis' table, on page 671 of the JOURNAL for 1892. The saving per day for 310 I.H.P. of main engines would then be

2.34 tons of 2,240 pounds. The coal per day from the tables is 15.12 tons at 8 knots, so that the saving would be 15½ per cent., while the weight of the extra machinery to do this is only 5¼ per cent. of the total coal carried, or 380 tons. On an expenditure of 15.12 tons per day she could steam 25.13 days on 380 tons, while with the expenditure reduced to 12.78 tons per day the 360 tons would allow 28.17 steaming days, both at 8 knots, or 3 days, equal to 576 miles, more with the reduced coal supply and additional machinery, This is on the supposition of a saving on the main engines alone. If we add the saving due to working the air, feed and bilge pumps off the main engine when cruising the saving will be appreciably greater. The power required to work the auxiliaries is given as 80.5 I.H.P. If the air, feed and bilge pumps formed the same proportion of this as on the *Charleston*, their aggregate power would be about 34 I.H.P. The power required when working smaller pumps by the main engine would not be as great as this; but still assuming it to be 34, and that we only gain by working them by an engine which develops a horse power on 1.9 pounds of coal instead of 4 pounds, the saving would amount to 1,714 pounds per 24 hours, equal to 0.765 ton. Adding this to our previous saving, we find the total saving to be 3.10 tons per day, making the total expenditure 12.02 tons instead of 15.12, making the total saving about 20 per cent., and still further increasing the radius of action and decreasing the cost of the vessel as a coal consumer thoughout her life.

Robert Forsyth, Esq., Associate.—Mr. Hollis speaks of the value of jacketing the cylinder heads as well as the barrel, to prevent internal condensation.

The late Mr. John Elder, in the early '60's, jacketed cylinder ends, barrels and heads of compound engines. At that time, 40 pounds per square inch was the boiler pressure. And in 1864, when steam pressure was increased to 75 pounds per square inch, he added the additional system of supplying steam to jacket the pistons of vertical engines. The steam was supplied through a hollow plunger, on the under side of the piston, working through stuffing boxes into a barrel arranged with

glass water tube, drain pipes and steam supply, and, although it was his practice so to fit new engines, it seldom was kept long in use, owing to the breaking of the plungers, caused by the main guide shoes wearing on a long run, the piston getting slack in the cylinder, and engine knocking. The plunger thus became the guide to the piston and broke very soon. When broken it was seldom renewed.

Jacketing the cylinder heads of vertical engines has developed some wonderful rigs, and the most common is to find that the steam inlet and outlet are so arranged that there is always a layer of water from $\frac{1}{2}$ inch to 2 inches inside the head.

About thirteen years ago some of the Austrian Lloyd's steamers had their cylinders jacketed by the hot gases from the smoke stack, circulation being caused by a small fan, but the deposit of soot in the casings and difficulty of cleaning them did not warrant the continuance of the device.

In regard to the use of live steam coils or reheaters in the receivers, their application leads to great bulk, and if they leak it is difficult to get at the parts for repair. Their use in stationary work has not resulted in the economy expected.

It would seem to me that the most economical way to run large triple-expansion engines at *very low powers* would be to arrange the L.P. engine forward and disconnect it, using the H.P. engine and the I.P. engine as a compound engine; but no economy on the main engines will make up for the waste arising from the use of numerous auxiliaries. During the early period of my experience at sea, we always had the sanitary pump, wash-deck pump and feed and bilge pump attached to main engine, and I can positively say that the deck and sanitary pumps so driven cost more money for repairs than would have supplied them with steam many times over.

Passed Assistant Engineer T. W. Kinkaid, U. S. N.—I think Mr. Hollis scores a good point in demonstrating the inadvisability of jacketing the barrel of the L.P. cylinder. I would go a step farther and not jacket even the heads of the cylinder, for the reason that the jacket must pass a good deal of

heat to the condenser, while the better plan of using a nest of tubes in the L.P. receiver would secure benefit from all the heat transmitted through the walls of this tubing.

The idea of coating the walls of the clearance spaces with a non-conducting material is excellent. In 1885, or thereabouts, in a correspondence with Chief Engineer Isherwood, I suggested the profuse lubrication of steam cylinders; that is, instead of injecting oil into the entering steam by drops, I proposed to put it in by pints and quarts and remove the oil from the exhaust steam by the best means available. The objects desired were two: coating the surfaces of cylinder and clearance space with a non-conducting film of oil and thereby reducing cylinder condensation; and the incidental advantage of excellent lubrication. The oil not only has a low conducting power, but a low specific heat, and would warm up to the cylinder temperature with an insignificant expenditure of heat.

The modern centrifugal eliminator now furnishes an excellent means of recovering the lubricating oil from exhaust, so that the same oil can be pumped into the steam chest over and over again.

An experiment essentially identical with the one proposed by me was carried out the very next year in England, by Mr. Donkin, I believe, and I understand that the results were very satisfactory.

One of the famous torpedo-boat builders of England has tried the use of a turn of copper tubing in the clearance space of a cylinder with marked benefit. Steam circulated continuously through the coil, and greatly reduced the initial condensation.

Mr. Hollis has certainly not exaggerated the difficulty of obtaining complete and reliable data for such calculations as he has undertaken. The accompanying table is a suggestion as to the best means of keeping the daily coal account of a modern steamship. It assumes the preparation of speed-and-water curves, or, rather, revolutions-and-water curves for each engine in the table. Of course, the making of these curves wonld involve at least a week's experimentation with the ship, each engine being exhausted separately to the condenser for a few hours at three

	Revolutions per minute.	Pounds water consumed per hour (taken from revolutions and water curve).	Water for N engines per hour.	Coal per pound of feed water per hour.	Coal for n engines one hour.	Number of hours in use.	Coal for h hours.	Water for h hours.
One main engine	R_1	W_1	n_1W_1	c	n_1W_1c	h_1	$n_1W_1ch_1$	$n_1W_1h_1$
One air pump	R_2	W_2	n_2W_2	c	n_2W_2c	h_2	$n_2W_2ch_2$	$n_2W_2h_2$
One circulating pump	R_3	W_3	n_3W_3	c	n_3W_3c	h_3	$n_3W_3ch_3$	$n_3W_3h_3$
One bilge pump	R_4	W_4	n_4W_4	c	n_4W_4c	h_4	$n_4W_4ch_4$	$n_4W_4h_4$
One feed pump	R_5	W_5	n_5W_5	c	n_5W_5c	h_5	$n_5W_5ch_5$	$n_5W_5h_5$
One blowing engine	R_6	W_6	n_6W_6	c	n_6W_6c	h_6	$n_6W_6ch_6$	$n_6W_6h_6$
Evaporators { On condenser								
Evaporators { On distiller	. . .	W_7	n_7W_7	c	n_7W_7c	h_7	$n_7W_7ch_7$	$n_7W_7h_7$
Heating ship { Cold weather								
Heating ship { Mod. cold "	. . .	W_8	W_8	c	W_8c	h_8	W_8ch_8	W_8h_8
Heating ship { Mild "								
Steering engine { Smooth sea								
Steering engine { Moderate sea	. . .	W_9	W_9	c	W_9c	h_9	W_9ch_9	W_9h_9
Steering engine { Rough sea								
							Total coal.	Total feed.

c = total coal ÷ total feed.

different speeds. The work could be carried on day and night.

I am aware that the compilation of these daily tables would add considerably to the already large amount of clerical work of the Engineer Department, but I suggest that the daily working out of indicator cards might be abandoned in favor of the more profitable work on the table of coal expenditure.

I think it might be a good idea to fit counters to the feed pumps, in order that the latter might do service as water meters, and thus afford a check on the total feed as arrived at in the table. Doubtless it would be practicable to calibrate a feed pump by operating it on the feed tank under proper conditions.

Chief Engineer Geo. W. Baird, U. S. N.—There is no subject which demands our attention more, at present, than that which Mr. Hollis has presented in his paper on economical speed. The unloading of the main engine of its air, circulating and feed pumps is of comparatively recent origin; it was forced upon us by the dangers incident to the "racing" and by the increased piston speed; but not with a view to increased economy. Up to 1870 the Government allowed us considerable sums of money for experiment, and, to that date, we had obtained experimental data enough to enable us to separate the losses, and determine with considerable accuracy where the heat was utilized and where lost. This was by no means a rough approximation. Since that period we have had neither men nor money to prosecute research in the science.

As early as 1874 I was detailed to calculate and tabulate the horses power of our own ships, for comparison with those of other Navies and with vessels of the mercantile marine, and I was surprised to find the closeness of performance of like engines. For example, there was the *Lackawanna* and class, 10 vessels; the *Nipsic* and class, 3 vessels; the *Chippewa* and class, 19 vessels. While the engines of each class were alike, the hulls differed somewhat; those of the *Lackawanna* and class were built in pairs. The little difference in the success of the vessels was easily accounted for at that time. All of these engines were

loaded with the pumps. I once heard the distinguished Lenthall say that a shipbuilder "could not build two hulls exactly alike; he could not even build the two sides of the same ship alike;" and he cited the *Niagara* as an example; she could sail faster on one tack than on the other with everything else equal. So it seems that were steam engineering an exact science we could not even then predict the exact speed of a ship.

The power to work the auxiliaries varies immensely. In the matter of pumps alone, it must not be overlooked that duplex pumps shorten up their stroke immensely (increasing steam clearance) when they are slowed down. Blower engines, where directly connected and run at a very high speed, are very wasteful of steam; it is difficult to indicate them.

A motor, known as the Wise motor, was directly connected to a blower on board a ship about ten years ago, and was represented as very economical. It was essentially a series of vanes radiating from a central shaft, and having jets of steam impinging on the vanes; the whole inclosed in an iron casing. I led the exhaust to the distilling apparatus, condensed and trapped the exhaust, and found it required 1,220 pounds of water per horse power per hour. It was then thrown out and a Greenfield engine substituted; I added both inside and outside lap to the valve of this engine, effecting considerable cushion and an earlier cut-off, and, upon indicating it, found it required 55 pounds of water per H.P. per hour. A leather belt was used to transmit the power.

My belief is that the best way to run the blowers is by rope transmission; the ropes may be led around angles, in any direction, thus permitting the engine to be placed where it is accessible. In this way one engine may be used to run a number of blowers. The present system of boxing each blower engine in, or, if exposed, run in a cloud of ashes, does not seem to me to be the best practice. On board the *Dolphin* we have two principal exhaust fans; they are connected to exhaust the air from the sleeping apartments, and force it into the fire room. These fans are run, ordinarily, from 500 to 600 revolutions a minute, which displaces from 1,200 to 1,500 cubic feet of air per

hour *per capita*, and at that speed they consume a mean of 226.41 pounds of steam per hour; the air and circulating pumps of the auxiliary condenser consume 220.75 pounds per hour; the dynamo 907.84 pounds per hour. These figures I obtained by trapping the condensed water and measuring it in the feed tank, and making a correction for temperature, all being referred to 100°. They are about a fair average; the purpose of this measurement was to ascertain the amount of coal to charge to each duty.

The most difficult auxiliary to standardize is the dynamo.

It may not be out of place to state here the difficulties encountered in this art, and some of the causes for the wide differences in electric economy. It is imperative to employ a uniform speed of engine and dynamo to obtain a steady light. Cut-off governors and high-speed engines are universally employed on shipboard for this purpose. As it is the custom to carry less pressure in port than at sea, it will at once be seen that the electric light companies (who give a guarantee of every plant) will invariably want to furnish a larger engine than we would consider right, in order to secure the power at the lower pressure. At sea, during the later hours of the night, there is less light used, but the boiler pressure and the vacuum in the condenser are kept up, and with the light load, the dynamo engine cuts-off shorter, increases its cylinder condensation, and diminishes its economy.

It is now well known that the higher the voltage used, the greater will be the economy of the output. It was my fortune to install the first incandescent electric lighting plant ever used on a Government steamer. The dynamo was wound for 51 volts, and there were 120 lamps of 8 candles power in the installation. The lamps required 38 watts, or $4\frac{3}{4}$ watts per c. p. The plant was considered a great success. I found that the number of lamps used, when the ship was at work, never exceeded 50, and rarely exceeded 48. This is a "load factor" of 40—that is, it is 40 per cent. of the load.

My next plant was 138 lamps of 10 candles power each, and by using 110 volts I increased the economy considerably. The

lamps were for 3.8 watts per candle power, or 38 watts each. It is here seen that I obtained an increased illumination of

$$\left(\frac{10-8}{10}\times 100=\right) \text{ 20 per cent.}$$

for the same electrical energy. The load factor was, as before, about 40 per cent.

The *Dolphin* (to which vessel I am now attached) has a compound wound dynamo, run directly by a pair of 5×3 engines. It is wound for 80 volts, and is figured for a total output of 9,600 watts. There are 150 lamps of 64 watts each (16 c. p.). The load varies between 40 and 60 ampères, and the pressure from 65 to 90 volts. The mean load factor is about 33 per cent.

The steam pressure is kept at 65 pounds at the dynamo engine; its actual speed is usually a little above 650 revolutions a minute; it is designed to cut off at one-quarter when below the speed of 750, so that its I.H.P. is about 12. The power of the engine driving the first dynamo was 5.85; that of the second was 4.84. Reducing all the lamps to 16 c. p., the plants will compare as follows: (1) 51-volt machine, 5 lamps per I.H.P.; (2) 110-volt machine, $6\frac{1}{2}$ lamps per I.H.P.; (3) 80-volt machine, $4\frac{1}{6}$ lamps per I.H.P.

The mean of these will probably be the correct unit on which to base the power to light a ship. This, it must be remembered, is the economy at the mean load factor, and not the result of the best effort. The power considered is the indicated horses power developed in the cylinder.

The cost per I.H.P. of the *Dolphin's* dynamo engine is $\left(\frac{907.84}{12}=\right)$ 75.8 pounds of water, assuming 12 H.P. to be correct. If the power is less, then the economy is even worse. Small plants are never so economical as large ones.

The load on a circulating pump, at a given speed, is nearly constant, and it is easy to arrive at the power to drive them.

The following table I constructed from the direct-acting circulating pump of the *Albatross;* its cylinders are 14 steam, 16 water, and 14 stroke. The steam cylinder is much too large. At 10 knots speed the main engine developed 457 horses power; the

air and feed pumps were worked from the main engine. At this power of the main engine, we required to run the circulating pump 80 strokes a minute, which required 5.84 horses power, or 1.27 per cent. of the indicated power of the main engine.

CIRCULATING PUMP OF U. S. S. *ALBATROSS.*

Revolutions per minute.	I.H.P. of steam cylinder.	Revolutions per minute.	I.H.P. of steam cylinder.
10	0.30	60	3.00
20	0.60	70	4.20
30	1.00	80	5.84
40	1.45	90	7.60
50	2.00		

The *Dolphin* has a direct-acting circulating pump engine which, by beam connections, drives the two air pumps. The horses power to work them is as follows:

Main engine	1,705	2,029.5	1,998	2,145	1,623
Air and circulating pump	66	40.46	40	41	25
Auxiliaries	54	48.04	50	67.4	.....
Total	1,825	2,118.00	2,088	2,253.4	1,648
Power required to work the air and circulating pumps in percentage of the aggregate H.P.,	3.62	1.91	1.92	1.81	1.52
Ratio of air-pump displacement to that of L. P. piston at the respective speeds	1 : 67.6	1 : 68.4	1 : 56.9	1 : 56.9	1 : 66.9

An alteration was made in the pumps after the first trial, so the first test should, very properly, be rejected; this leaves the power to work these pumps less than two per cent.

Passed Assistant Engineer John L. Gow, U. S. Navy.—Mr. Hollis' reference to the formulæ of Chief Engineer Lowe reminds me that I had occasion to test it shortly after it was published, when I was crusing in the Pacific, where long distances and limited coal supply frequently furnished opportunity for the solution of a similar problem.

I found that, when working with data culled from the log book, when the circumstances were about the same, I got practically the same result; but I found that if I took my data from

widely separate parts of the book, the results varied widely. The method of using a tangent to the curve of performance to determine the minimum value of coal divided by distance run suggested itself to me then, and later, when an instructor at the Naval Academy, I taught it to the cadets as a sort of corollary to Sennett on speed curves and coal endurances.

Mr. Hollis mentions the change of position which this curve assumes for the various variable conditions of wind, sea, draught, condition of bottom, condition of machinery, quality of coal, etc., and I have long thought that, had we such curves accurately determined for our vessels, they could be made exceedingly useful, not only for purposes of future design, but for everyday, practical use when cruising at sea. For instance, suppose such a curve to be in the possession of the Chief Engineer of a vessel, and a tin or paste board template to be made from it; then suppose, after a day's steaming, at any speed, that the speed and coal per hour would plat at some point—say x, Fig. 2—then placing the template at a and x, a new curve could be drawn through these two points, which would be, very closely, the vessel's curve of performance for the conditions of weather, coal, etc., that then existed. A tangent might then be drawn, and the most economical speed determined for the conditions existing at that time.

Another item obtainable from such a curve, and one constantly desired to be known by commanding officers, would be the probable speed for any change in the coal expenditure. But this is all theorizing. Practically, we could do all this had we reliable curves of performance, but without careful progressive trials, with data for lower speeds than any yet made, we cannot get the curve.

It seems to me, however, that it is a subject of sufficient importance to urge that our ships during their tentative period, before being sent on regular cruising duty, should be given trials that will give us their curve of performance accurately, so that they might thus be made of practical use. In other words, it is as important to know the range of the ship at all times as it is to know the range of her guns, and to wait till such data could be picked up or guessed at is as poor policy as it would be to delay ranging her guns till in actual service.

For the purpose of comparing the efficiency of vessels, the tangent of the angle at maximum endurance would constitute a most excellent criterion, that vessel being the best in all respects, as to design and condition, where the above quantity was the least.

Passed Assistant Engineer Hollis.—The interest taken in the above paper on economy at low speeds is very gratifying, and I hope we may be able to obtain data from ships now in commission to aid in solving this question of speeds for future ships. In regard to Mr. Isherwood's suggestion about collecting data from the steaming log. I tried his plan last fall. The coal consumption taken from the log for speeds from 6½ to 7½ knots, from 7½ to 8½ knots, and so on, up to the highest speeds, gave very unsatisfactory results. One ship, for instance, had the same endurance from 6 up to 14 knots. The reason is not far to seek. The same speed may be made under such different conditions of sea and weather, draught and cleanness of bottom, that the mean consumption of coal will be far out. The *Charleston* burnt on one voyage, with foul bottom, about 53 tons of coal per day, at 8.8 knots, which gave her an endurance of 2,995 knots; on another voyage, with clean bottom, she made 8.8 knots on 30 tons of coal a day, which made the endurance 5,293 knots. The mean of these two would be 4,144 knots, just 1,149 knots out in either case. I found the most satisfactory method was to take each voyage separately and work out the endurance from the average daily coal consumption and the average speed for the voyage. The displacement varied from day to day, but the mean result was as near the actual performance as could be obtained from the log.

THE CONTRACT TRIAL OF THE U. S. S. *MONTEREY*.

By Passed Assistant Engineer Howard Gage and Assistant Engineer Emil Theiss, U. S. Navy.

The *Monterey* is a twin-screw, steel, armored vessel of the monitor type, and was built by the Union Iron Works, of San Francisco, Cal., the contract price being $1,628,950. She is the last but two of the ships recently added to the Navy whose contract calls for a specified indicated horse power to be developed by the propelling machinery. The requirement in the present case is for an I.H.P. of 5,400 for the main engines and the air and circulating pumps. For every horse power in excess of this figure a premium of $100 is to be paid, while a penalty of like amount is to be exacted for every I.H.P. by which the power developed falls short.

The feature of special interest in connection with the machinery of this vessel is the coil boilers, which are used in conjunction with two single-ended Scotch boilers. It became apparent in the course of making the designs that, to answer the demands made for weight of armor and armament, great reductions would have to be made elsewhere. It was proposed, therefore, by the Bureau of Steam Engineering to try the experiment of coil boilers in battery. Competition was invited, and the record of the trials which resulted in the selection of the Ward boiler may be found in Vol. II, No. 3, p. 414 of the Journal, as well as in the report of the Engineer-in-Chief for the year 1890. The coil boilers were contracted for to give 4,200 I.H.P., the remainder, 1,200 I.H.P., to be furnished by the Scotch boilers.

The steel-belt armor extends the entire length of the vessel, from 2 feet 6 inches above the cruising load-water line to 2 feet below. For a length of 118 feet the armor is 13 inches thick from the top of the armor to 9 inches above the cruising water

line; thence tapering to 5 inches at the armor shelf. For a length of 19 feet immediately forward and aft of the 13-inch belt the armor is 8 inches thick to 9 inches above the cruising water line, tapering to 4 inches at the shelf. Forward and abaft the 8-inch armor the belt is 6 inches thick, tapering to 4 inches at the shelf.

The main battery consists of two 12-inch (46-ton) guns and two 10-inch (27-ton) guns. These guns are mounted in two barbette turrets—the 12-inch guns forward, the 10-inch aft. The secondary battery consists of six 6-pounders, mounted on the superstructure deck; two 1-pounders on the bridge, and two in the military top.

The barbette armor of the forward turret is 13 inches thick, the revolving armor being 8 inches thick. These thicknesses for the after turret are 11½ and 7½ inches. The conning tower is of 10-inch steel, connected with the armored deck by a 16-inch tube 6 inches thick. The smokepipe and the fire-room ventilator are armored to 4 feet above the armored deck by steel 6 inches thick. The armored deck consists of two courses of an aggregate thickness from the stem to the after side of the after barbette of 3 inches, and aft of that of 2 inches.

There is a military mast fitted with one top for machine guns, and two booms for handling boats.

The double bottom extends from frame No. 5 to frame No. 77, and is divided into 107 separate water-tight compartments. Forward of the double bottom are two compartments extending up to the ramplate flat. Aft of the double bottom are also two compartments extending up to the steering engine flat. The total number of water-tight compartments in the ship is 223.

The coal bunkers, capacity about 200 tons of anthracite coal, are divided into six separate compartments, three on each side, fore and aft, in wake of the engine and boiler compartments, and two athwartships, at the forward end of the boiler rooms, divided by the midship longitudinal bulkhead. The fore and aft bunkers do not extend out to the ship's side, from which they are cut-off by the wing passage bulkheads. The spaces outboard above the level of the berth deck form passageways, and the

spaces below are utilized for leading the ventilating ducts fore and aft.

The compartments in the double bottom are divided by water-tight transverse frames, into nineteen groups. In the port midship one of each group communication is made by a stop valve with the secondary drain pipe, a four-inch pipe, running on the port side of the keelson from compartments No. 1 to No. 108, with a stop-valve in each of these compartments. The ship is submerged to her fighting trim by admitting water into the double bottom. Nine flushing Kingstons are provided for this purpose, located at the side of the ship on the berth deck and in the passageways, and air cocks are fitted for each compartment. The water is admitted to the different compartments by sluice gates. The water-tight compartments above the double bottom communicate through sluice valves with those compartments in which are located the main drainage cisterns. These latter are fitted with a strainer, a stop and a non-return valve, and communicate with the 11-inch main drain pipe running fore-and-aft on the starboard side of the keelson. These cisterns are seven in number. There is one in the after part of each boiler room and one in the forward part of each engine room. Those in the engine room communicate through a sluice gate in the midship keelson, and from the port cistern a short section of pipe leads forward, terminating in the drainage cistern in the port boiler room. Connection is established between the main and secondary drainage systems by three sections of pipe with stop-cocks connecting the mains. All drain and stop valves, main drain valves, sluice gates, sluice valves, and sounding tubes, excepting those located in the machinery spaces, are operated from the berth deck.

The ship is ventilated by two Sturtevant blowers of 10,000 cubic feet capacity per minute. Damper valves are arranged so that fresh air can be supplied to the different compartments, or foul air exhausted therefrom. Emergency ducts are provided, running through the thwartship coal-bunkers to the boiler rooms, so that with battle hatches closed and the ventilators shot away,

the air supply may be drawn down the armored ventilator supplying the fire-room blowers.

The steering is done by hydraulic power, pressure being created ordinarily by a three-cylinder plunger pump in the port engine room, or by the pumping engines supplying the power for turning the turrets and working the guns. Incidentally, it may be stated that hydraulic power is also used for operating the battle hatches, in addition to hand gear. The steering gear consists of two hydraulic cylinders, with pistons and rods directly operating a tiller-head, from which connecting rods transmit the motion to the rudder-head. The hydraulic distributing valve is worked by a lever receiving its motion from a nut traveling on a screw fixed endwise, and revolved by steel-wire ropes transmitting the motion from the hand wheels. The lever actuating the valve is a float lever, and the valve is closed as fast as opened by a system of levers receiving motion from the tiller-head. Steering wheels are provided in the pilot house, the conning tower and the steering-gear room. Here, also, are located wheels for steering by hand, a clutch sliding on a feather serving to throw the hand power in and the hydraulic-valve gear out. The hydraulic cylinders and connections remain in gear and act as buffers, a by-pass valve being provided to connect the pipes to the two ends of the cylinders.

Fresh and salt water for flushing purposes, bathtubs, pantries, &c., is distributed over the ship by gravity. The fresh water is pumped by a hand pump on the berth deck from the water tanks in the hold to a distributing tank on the superstructure deck, from which a system of piping conveys it to the different parts of the ship. Side by side with the fresh-water tank is another for salt water. The circulating pump for the distiller discharges into this, either directly or after passing the water through the distiller. Connection is also made with the fire main, so that any of the steam or hand pumps supplying the latter may be used for flushing purposes, and a valve is interposed between the branch from the fire main and the salt-water tank, so that the flushing system can be put under pressure should obstructions collect.

HULL.

Material	Mild steel.
Where built	Union Iron Works.
When built	1890–1893.
Length between perpendiculars	250 feet ½ inch.
Length on L.W.L.	256 feet.
Length over all (including rudder)	262 feet 4 inches.
Beam, extreme	59 feet ½ inch.
Beam at L.W.L.	59 feet ½ inch.
Ratio of length to beam	4.34
Depth in hold from top of main-deck beams to top of floors	14 feet 6 inches.
Draught forward, sea-going trim	14 feet 6 inches.
Draught aft, sea-going trim	14 feet 6 inches.
Displacement, sea-going trim (load draught)	4,027 tons.
Displacement, fighting trim	4,350 tons.
Displacement per inch at L.W.L.	26.9 tons.
Area of immersed midship section, sea-going trim	775 square feet.
Area of immersed midship section, fighting trim	830 square feet.
Area of L.W.L. plane	11,263 square feet.
Center of gravity of L.W.L. plane forward of midship section, Fr. 43	0.05 feet.
Center of buoyancy above bottom of keel	8.1 feet.
Center of buoyancy forward of midship section	0.635 feet.
Transverse metacenter above center of buoyancy	17.8 feet.
Longitudinal metacenter above center of buoyancy	292 feet.
Coefficient of fineness on extreme dimensions	0.643
Coefficient of fineness on midship section	0.710
Coefficient of fineness of L.W.L.	0.745
Wetted surface	19,250 square feet.

ENGINES.

There are two vertical, direct-acting, triple-expansion engines placed abreast of each other in separate water-tight compartments. There is a water-tight door in the central longitudinal bulkhead on a level with the working platform. The cylinders are not jacketed, the H.P. cylinder being furnished with a working liner.

The valves are all of the single-ported piston type, one for the high and two each for the intermediate and low-pressure cylinders. As originally fitted, the valves were provided with float rings and packing rings. It was found on one of the preliminery runs that there was excessive cushioning in the H.P. and I.P. cylinders, necessitating considerable change of the exhaust laps. The valve bodies were retained, but for the float

rings and packing rings were substituted solid cast-iron rings split longitudinally and bolted together, to allow of insertion of liners to make up for wear. The valve-gear is of the Stephenson type with double bar links. The I.P. and L.P. cylinders, having two valves, have valve stem crossheads. These are fitted inside the valve chests and permit of longer eccentric rods. There are no cut-off valves fitted. By means of the slot in end of arm on reversing shaft, however, an independent adjustment of cut-off for each cylinder, varying from 0.5 to 0.7, of the stroke is provided.

Starting valves are fitted to all cylinders with suitable handling gear operated from the working platform. The reversing engine consists of a vertical steam cylinder, 12 by 12 inches, and a hydraulic controlling cylinder of 6 inches diameter, fitted with a floating lever mechanism. A small oil pump is provided for reversing by hand. The throttle is of the gridiron type, operated by means of a vertical shaft with hand-wheel and suitable connections. The piston valves are balanced; that for the H.P. cylinder by connections with the condenser above the balance piston, those for the I.P. and L.P. cylinders by steam pressure acting against the lower faces of balance pistons, which latter are fitted to work in the lower valve chest covers. These latter balance pistons did not work satisfactorily owing to difficulty experienced in keeping their cylinders clear of water. They were removed prior to the official trial, and have not been again put in place.

The main pistons are cast steel, cone shaped, and fitted with two packing rings each $\frac{5}{8}$ inch wide by $\frac{3}{4}$ inch deep. The piston and connecting rods are of forged steel and interchangeable among the cylinders. The crank-pin and wrist-pin brasses, as well as the eccentric straps, are lined with Parson's white metal. The crossheads are of forged steel, bolted to composition shoes fitted with phosphor bronze slippers faced with Parson's metal. The engine frames are of the inverted Y type, two to each cylinder, fitted with steel plate stiffening pieces at their bases. They are faced with cast-iron crosshead guides, cored for water circulation. The bed plates are of cast steel, each in three sec-

tions securely bolted together. The crank-shaft bearings are separate, of phosphor bronze lined with Parson's metal.

There are no outside exhaust pipes between the cylinders. The valve chests of the high and intermediate pressure cylinders are bolted together. The exhaust from the I.P. cylinder passes around the latter, through a passage which forms part of the cylinder casting and into the L.P. valve chest, which is bolted to flanges on the I.P. cylinder.

Number of cylinders (each engine)		3.
Diameter of H.P. cylinder	Both engines	27.08 inches.
Diameter of I.P. cylinder	Both engines	41.09 inches.
Diameter of L.P. cylinder	Both engines	64.10 inches.
Stroke of all pistons, excepting starboard L.P.		30 inches.
Stroke of starboard L.P. piston		30 1/32 inches.
Diameter of H.P. valve (one for each cylinder)		14 inches.
Diameter of I.P. valves (two for each cylinder)		14 inches.
Diameter of L.P. valves (two for each cylinder)		20 inches.
Area of H.P. steam ports for maximum opening	Top	65.79 square inches.
	Bottom	51.40 square inches.
Area of I.P. steam ports for maximum opening (two valves)	Top	131.58 square inches.
	Bottom	102.8 square inches.
Area of L.P. steam ports for maximum opening (two valves)	Top	209.29 square inches.
	Bottom	203.18 square inches.
Area of H.P. exhaust port for maximum opening		82.24 square inches.
Area of I.P. exhaust ports for maximum opening (two valves)		164.48 square inches.
Area of L.P. exhaust ports for maximum opening (two valves)		317.76 square inches.
Area of main steam pipe (8½ inches diameter)		56.745 square inches.
Area of main exhaust pipe (20 inches diameter)		314.16 square inches.
Volume swept by H.P. piston per stroke		9.828 cubic feet.
Volume swept by I.P. piston per stroke		22.851 cubic feet.
Volume swept by L.P. piston per stroke	Starboard	55.861 cubic feet.
	Port	55 855 cubic feet.
Clearance of H.P. cylinder, per cent	Starboard	13.7
	Port	13.1
Clearance of H.P. cylinder in inches of stroke	Starboard	4.11
	Port	3.93
Clearance of I.P. cylinder, per cent	Starboard	10.7
	Port	9.3
Clearance of I.P. cylinder in inches of stroke	Starboard	3.21
	Port	2.79
Clearance of L.P. cylinder, per cent	Starboard	7.0
	Port	6.8
Clearance of L.P. cylinder in inches of stroke	Starboard	2.10
	Port	2.04

Ratio of net area of H.P. to I.P. pistons	2.302
Ratio of net area of I.P. to L.P. pistons	2 434
Ratio of net area of H.P. to L.P. pistons	5.589
Diameter of piston rods	5 inches.
Length of connecting rods between centers	60 inches.
Diameter of connecting rods	5 and 7¼ x 5 inches.

Shafting and Bearings.—All the crank, line and propeller shafting is of forged steel, with an axial hole through shafting and crank pins. The crank shaft for each engine is in three interchangeable sections, bolted together by bolts with head let into flanges. The sequence of cranks for head-going motion is H.P., I.P., L.P. The line shafting is in three sections, supported on five bearings. The section next abaft the crank shaft is not rigidly coupled to the succeeding section, but there is ⅜-inch play between the coupling flanges, the bolts being fast in the after flange of the short section and free in the forward flange of the succeeding length, to which a centrifugal oiler is fitted for lubricating the bolts. The propeller shafts are cased with composition from inboard of the stern-tube stuffing box to the propellers, a length of 27 feet 8 inches. The inboard ends are fitted with detachable couplings of forged steel, secured to the shaft by three feathers and a stud and washer. The thrust bearing is of the collar-thrust type, and is lined with white metal. The pedestal is formed into an oil trough, with an adjustable bearing at each end for taking the weight of the shaft, and stuffing boxes for confining the oil.

Diameter of crank shaft	11 inches.
Length of each section (three)	6 feet 9 inches.
Diameter of crank-pin	11 inches.
Length of crank-pin	15 inches.
Width of crank-web	14 inches.
Thickness of crank-web	4½ inches.
Diameter of flanges	21 inches.
Thickness of flanges	3 inches.
Number of coupling bolts (one flange)	6.
Diameter of coupling bolts	2⅞ inches.
Diameter of hole in shaft	4 inches.
Diameter of hole in crank-pin	4 inches.
Total length of crank-shaft bearings	6 feet.
Diameter of line-shaft	10 inches.

Length of line-shaft, total (3 sections)..... 37 feet 5⅞ inches.
Diameter of hole in line-shaft..... 4 inches.
Diameter of flanges..... 20 inches.
Thickness of flanges..... 2½ inches.
Number of bolts in one flange..... 6.
Diameter of bolts..... 2¾ inches.
Diameter of thrust shaft..... 10 inches.
Length of thrust shaft..... 12 feet 11 inches.
Diameter of hole in thrust shaft..... 4 inches.
Diameter of flanges..... 20 inches and 21 inches.
Thickness of flanges..... 2½ inches and 3 inches.
Number of bolts in each flange..... 6.
Diameter of bolts..... 2¾ inches.
Number of thrust collars..... 12.
Diameter of thrust collars..... 13¾ inches.
Diameter of shaft in thrust bearing..... 10½ inches.
Actual thrust surface..... 725.67 square inches.
Thickness of collars..... 1½ inches.
Space between collars..... 1⅞ inches.
Diameter of propeller shafts..... 10 inches.
Length of propeller shafts, total (1 section)..... 33 feet 9 inches.
Diameter of hole..... 4 inches.
Thickness of composition sleeve... { at bearings..... ¾ inch. / elsewhere..... ½ inch.
Length of sleeve..... 27 feet 8 inches.
Length of bearing in stern tube (2 sections)..... 6 feet 5⅜ inches.
Length of bracket bearings..... 2 feet 7 inches.

Condensers.—The condensers are of cast brass, each in five sections, including the heads, bolted together. The water circulates through the tubes. There being no auxiliary condenser, the main condensers are to be used as such, and a gridiron valve is fitted at the base of the exhaust pipe from the main engines. The bonnets of the water chests, containing the inlet and outlet openings for the circulating water, can be taken off without disturbing the pipe connections.

Length over all..... 9 feet 11½ inches.
Diameter of shell, inside..... 5 feet 0 inch.
Thickness of shell..... ⅜ inch.
Length of tubes exposed..... 8 feet 0 inch.
Diameter of tubes, outside..... ⅝ inch.
Thickness of tube, B.W.G., No. 20..... .035 inch.
Number of tubes, one condenser..... 2,944.
Cooling surface, one condenser..... 3,853.7 square feet.
Cooling surface of two condensers to H.S. of boilers..... 1 to 1.905

Air, Circulating and Hot-well Pumps.—There is one double, vertical, single-acting air-pump for each main engine, built by the Dow Pump Works and fitted with Dow's valve gear and governor. The pumps are connected by a beam pivoted at its center. The steam cylinders are two in number, placed directly above the pumps, the piston and pump rods forming a continuous length. The beam receives its motion through links swinging from a crosshead on the pump and piston rods. It is the intention to use only one steam cylinder at a time, the other being held in reserve.

The air pump discharges the air, water and vapor drawn from the condenser into the hot-well, from which the air and vapor find their way into the escape pipe, while the water is drawn off by a small Dow light-service pump known as the *hot-well pump*, and discharged into the feed tanks, which are located in the boiler rooms.

There is a centrifugal circulating pump for each condenser, arranged to draw either from overboard or from the bilge, and to discharge either into the condenser or overboard. It is driven by an inclined, inverted, compound engine.

AIR PUMPS.

Diameter of steam cylinders (two)	12 inches.
Diameter of water cylinders (two)	20 inches.
Stroke	20 inches.
Diameter of piston rod	2 inches.
Diameter of pump rod	3 inches.
Number of foot valves (rubber) each pump cylinder	10.
Diameter of foot, bucket and delivery valves	5 inches.
Number of bucket valves	8.
Number of delivery valves	8.
Ratio of volume swept by L.P. piston per stroke to that of air-pump buckets, per stroke	7·77

HOT-WELL PUMPS.

Diameter of steam cylinder	4½ inches.
Diameter of water cylinder	7 inches.
Stroke	8 inches.
Capacity per stroke	1.66 gallons.

CIRCULATING PUMPS.

Diameter of steam cylinders	5½ inches and 11 inches.
Stroke	6 inches.

Diameter of runner ..... 28 inches.
Width of runner ..... at hub, 5¾ inches ; at tip, 2 inches.
Diameter of runner shaft ..... 3 inches.
Diameter of suction ..... 11 inches.
Diameter of discharge ..... 11 inches.
Capacity of one pump per minute ..... ,000 gallons.

Screw Propellers.—The propellers are of manganese bronze, three bladed, adjustable, and with axially and radially expanding pitches.

The port propeller is left-handed, and the starboard right-handed. The hubs are spherical, and fitted with conical tail-pieces.

Number of blades ..... 3.
Diameter of propeller ..... 10 feet 2 inches.
Diameter of hub ..... 3 feet.
Length of hub ..... 2 feet, 4¼ inches.
Pitch at hub—entering, 11 feet; leaving, 13 feet; mean ..... 12 feet.
Pitch at tip—entering, 11 feet 6 inches; leaving, 13 feet 6 inches; mean ..... 12 feet 6 inches.
Pitch as set on trial, entering, mean ..... 10 feet 8 inches
Pitch as set on trial, leaving, mean ..... 12 feet 8 inches } Mean... 11 feet 8 inches.
Helicoidal area, for one screw ..... 38.88 square feet.
Projected area, for one screw ..... 29.07 square feet.
Disc area, for one screw ..... 81.18 square feet.
Ratio, pitch to diameter 1.138
Ratio, helicoidal area to disc area ..... .475
Ratio, projected area to disc area ..... .355
Ratio, disc area of two propellers to area of immersed midship section ..... .209
Immersion of center of propeller at draught of 15 feet 5½ inches aft... 9 feet 8½ inches.

Boilers.—There are two single-ended, cylindrical, horizontal fire-tube boilers, and four Ward boilers, making six (6) in all. They are erected in two fore-and-aft water-tight compartments, separated by a midship longitudinal bulkhead. Each fire room contains one cylindrical and two Ward boilers.

In the *cylindrical boilers*, the plates, furnaces, tubes, rivets and braces are of steel, the grate bars of wrought iron. The longitudinal shell joints are treble-riveted with double-butt straps; the circumferential joints are lapped and double riveted.

In the Ward boilers, the coils and vertical, central drum are of wrought steel, the manifolds and heads of central drum of cast steel, the grate bars of cast iron and the boiler casing of light

sheet iron, lined with fire brick and tiles; the top of the casing, also, is of light sheet iron.

The uptakes from all boilers discharge into a common smoke-pipe, oval in shape.

The following are some of the dimensions of one cylindrical boiler:

Length	10 feet 7 inches.
Diameter, outside	11 feet 2 inches.
Number of furnaces, corrugated	2.
Diameter of furnaces, outside	3 feet 10 inches.
Diameter of furnaces, inside	3 feet 6 inches.
Length of grate	6 feet $3\frac{3}{4}$ inches.
Outside diameter of tubes (steel)	$2\frac{1}{2}$ inches.
Length of tubes between tube sheets	7 feet $4\frac{7}{8}$ inches.
Spacing of tubes horizontally	$3\frac{1}{2}$ inches.
Spacing of tubes vertically	$3\frac{1}{4}$ inches.
Number of common tubes	212.
Number of stay tubes	70.
Thickness of common tubes, No. 12, B.W.G.	.109 inches.
Thickness of stay tubes, No. 6, B.W.G.	.203 inches.
Total number of tubes in one boiler	282.
Number of combustion chambers	2.
Depths of combustion chambers	27 and $28\frac{1}{4}$ inches.
Width of combustion chambers	4 feet $6\frac{1}{4}$ inches.
Height of combustion chamber above back end of grates	4 feet $2\frac{3}{8}$ inches.
Thickness of shell plates	$\frac{63}{64}$ inch.
Thickness of heads, Top plate	front, $\frac{7}{8}$ inch; back, $\frac{7}{8}$ inch.
Thickness of heads, Bottom plate	front, $\frac{7}{8}$ inch; back, $\frac{3}{4}$ inch.
Thickness of back tube sheet	$\frac{3}{4}$ inch.
Thickness of combustion-chamber plates	$\frac{1}{2}$ inch.
Thickness of furnaces	$\frac{13}{32}$ inch.
Diameter of rivets in longitudinal joints	$1\frac{1}{8}$ inches.
Diameter of rivets in circumferential joints	$1\frac{1}{8}$ inches.
Pitch of screw stays	7 inches.
Diameter of screw stays	$1\frac{3}{16}$ inches and $1\frac{3}{8}$ inches.
Number and diameter of through braces above tubes	8 of 2 inches.
Number and diameter of braces from head to back tube sheet	4 of $1\frac{3}{4}$ inches.
Number and diameter of diagonal braces	2 of $1\frac{3}{4}$ inches.
Number and diameter of braces around lower manhole	3 of 2 inches.
Tube-heating surface	1,252.72 square feet.
Plate-heating surface	199.82 square feet.
Total heating surface	1 452.54 square feet.
Grate surface	44 square feet.
Area through tubes	6.12 square feet.

Area over bridge walls, (two)	5.75 square feet.
Area through smoke duct (18x42 inches)	5.25 square feet.
Volume of furnaces and combustion chambers above grates	182 cubic feet.
Steam room, water 6 inches above tubes	147 5 cubic feet.
Water surface, water 6 inches above tubes	98 square feet.
Diameters of smoke pipe	10 feet by 6 feet.
Area of smoke pipe	51.72 square feet.
Height of smoke pipe above grate	61 feet 6 inches.
Diameter and number of safety valves in one case	2 of 3 inches.
Load on safety valves	160 lbs.
Diameter main boiler stop valve	5 inches.
Diameter auxiliary boiler stop valve	5 inches.
Test pressure	250 pounds.
Totals for two cylindrical boilers:	
Tube heating surface	2,505.44 square feet.
Plate-heating surface	399.64 square feet.
Total heating surface	2,905.08 square feet.
Grate surface	88 square feet.
Area through tubes	12.24 square feet.
Area over bridge walls	11.50 square feet.
Area through uptakes	10.50 square feet.
Area of water surface	186 square feet.
Volume of steam room	295.0 cubic feet.
Volume of furnaces and combustion chambers above grates	364 cubic feet.
Weight of boilers, with fittings and water, per 100 square feet of heating surface	1.27 tons.
Ratios:	
Tube H.S to G.S	28 47 to 1.
Plate H.S. to G.S	4.54 to 1.
Total H.S to G.S	33.01 to 1.
Area through tubes to G.S	0.139 to 1.
Area of water surface to G.S	2.114 to 1.
Steam room per square foot of G.S	3 352 to 1.
Volume of combustion chambers and furnaces above grate and bridge wall to G.S	4.137 to 1.

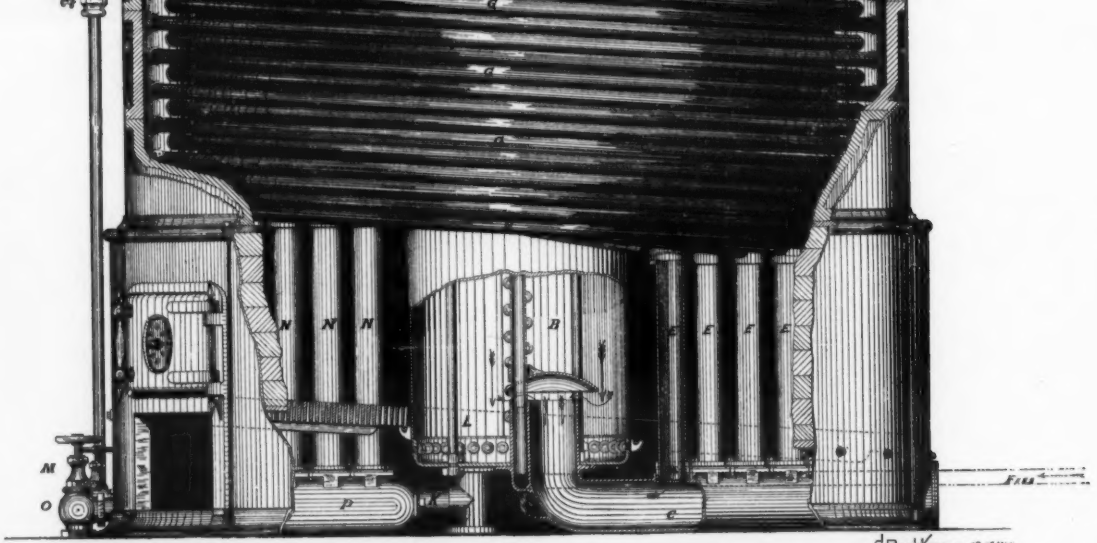
Ward boilers.—The Ward boilers used on the *Monterey* are of the type found described in Vol. II, No. 2, p. 177 of the JOURNAL. Their arrangement is sufficiently familiar to render only a general description necessary in connection with the accompanying illustration of a similar but smaller boiler. The coils are connected to the vertical manifolds by means of nipples with right and left-hand threads. The vertical manifolds are connected with the lower horizontal manifolds by 3 inch pipes

screwed into the vertical manifolds, and lipped and secured into stuffing boxes in the horizontal manifolds by glands made in halves, the flanges dovetailed together. The upper horizontal manifold has rectangular lipped extensions cast on its lower side. These fit into stuffing boxes in the tops of the vertical manifold and are held by glands made in halves. At both ends the stuffing box bolts are inserted in T-shaped slots, and for the upper manifold the flanges are extended to protect the bottom of the manifold from the impact of the flames. The upper horizontal manifold is connected to the vertical central drum by means of a shallow stuffing-box and gland. The packing material throughout is asbestos.

The central drum is provided with a cylindrical diaphragm near the bottom, to separate the upward from the downward currents of water, and its outside is cased with fire bricks $4\frac{1}{2}$ inches thick to a height of 24 inches above the grate, to protect it from the intense heat of the furnace. The boiler casing surrounding the furnace is of sheet iron $\frac{1}{8}$-inch thick. To prevent radiation, this casing, to the height of the top of the furnace is lined with fire bricks $4\frac{1}{2}$-inches thick. Above this the casing is of $\frac{3}{32}$-inch sheet iron lined with hollow backed tiles $1\frac{3}{4}$ inches thick. The cover is of $\frac{3}{32}$-inch sheet iron, made in sections for removal, and is not faced with any non-conducting material. The grate is annular, of cast iron, in 24 sections.

The following are some of the principal dimensions and proportions of this boiler:

Outer diameter of annular grate	10 feet 2 inches.
Inner diameter of annular grate	3 feet $0\frac{3}{4}$ inch.
Height of furnace	24 inches.
Outside diameter of casing at furnace	11 feet 1 inch.
Outside diameter of casing above furnace	10 feet 8 inches.
Height of casing from floor to cover (about)	8 feet 3 inches.
Height of central drum from floor (about)	11 feet 6 inches.
Length of central drum, outside	10 feet 9 inches.
Diameter of central drum, outside	28 inches.
Thickness of metal in drum	$\frac{5}{16}$ inch.
Number and size of through stays in drum	4 of $1\frac{1}{4}$ inches.
Height of boiler from floor to top of smoke duct (about)	12 feet 4 inches.
Dimensions of smoke duct, inside	2 feet 3 inches by 4 feet 6 inches.



WARD'S STEAM GENERATOR.

TOTAL { HEATING SURFACE 1881 / GRATE " 42 } SQUARE FEET

N24.

Diameter of stop valve ..... 5 inches.
Diameter and number of safety valves ..... 2 of 3 inches.
Test pressure ..... 300 pounds.
Number of vertical manifolds, nine on a side ..... 18.
Number of concentric coils ..... 18.
Number of tubes in concentric coils ..... 540.
Outside diameter of tubes ..... 1.9 inches.
Thickness of tubes ..... $\frac{1}{8}$ inch.
Number of coils for each pair of vertical manifolds ..... 30.
Heating surface, total ..... 2,970 square feet.
Grate surface ..... 73.81 square feet.
Volume of steam space ..... 18 cubic feet.
Volume of water space ..... 86 cubic feet.
Area of air passage around coils ..... 14 square feet.
Area of uptake ..... 10 square feet.
Total for four Ward boilers:
Heating surface, coils ..... 11,022 square feet.
Heating surface, vertical manifolds ..... 608 square feet.
Heating surface, central vertical drum ..... 250 square feet.
Heating surface, total ..... 11,880 square feet.
Grate surface ..... 295.24 square feet.
Volume of steam room ..... 72 cubic feet.
Volume of water to second gauge ..... 344 cubic feet.
Area of air passages around coils ..... 56 square feet.
Area of uptakes ..... 40 square feet.
Ratio, coil H.S. to G.S ..... 37.33 to 1.
Ratio, vertical manifold, H.S. to G.S ..... 2.051 to 1.
Ratio, central vertical drum, H.S. to G.S ..... 0.844 to 1.
Ratio, total H S. to G.S ..... 40.23 to 1.
Ratio, area around coils to G.S ..... 0.189 to 1.
Ratio, area of uptakes to G.S ..... 0.134 to 1.
Ratio, steam room to G.S ..... 0.243 to 1.
Weight of boilers, fittings and water, per 100 square feet of heating surface ..... 0.589 tons.
Totals for all boilers:
Tube-heating surface ..... 13,527.44 square feet.
Plate-heating surface ..... 1,257.64 square feet.
Total heating surface ..... 14,785.08 square feet.
Grate surface ..... 383.24 square feet.
Area through tubes ..... 12.24 square feet.
Area around coils ..... 56 square feet.
Area of uptakes ..... 52.5 square feet.
Area of smokepipe ..... 51.72 square feet.
Capacity of steam room ..... 367 cubic feet.
Ratios of all boilers:
Tube H.S to G.S 35.32

Plate H.S. to G S	3 28
Total H.S. to G.S.	38 59
Area of uptakes to G.S.	0.137
Area of smokepipe to G.S.	0.135
Steam room, per square foot of G.S.	0.958
Weight of boilers, fittings, water, and all fire-room weights, per 100 square feet of H. S.	1.489 tons.

Forced Draft.—The closed fire-room system of forced draft is used. Four fan blowers, two in the after part of each boiler room, draw the air down a cylindrical shaft, armored above the armor deck, and deliver directly into the fire rooms. The fans are driven by enclosed, compound, vertical, direct-acting engines.

Diameters of steam cylinders	5 inches and 8 inches.
Stroke	6 inches.
Diameter of fans	5 feet.
Width of fans at rim	9 inches.

Feed pumps.—In each boiler room, there are two vertical, single, double-acting, Dow pumps for main and auxiliary feed. The feed systems of the two boiler rooms are connected through gate valves at the midship bulkheads, but the auxiliary system is entirely distinct from the main feed system. The main feed pumps draw from the feed tanks only and deliver into the main pipes only. The auxiliary feed pumps draw from the feed tanks, from overboard, from the fire room bilges, and from the boilers, and deliver into the auxiliary feed pipe, the fire mains, fire plugs in fire rooms, and the pipe connecting the tanks of the hydraulic pumping engines. The dimensions of these pumps are: Steam cylinder, 10 inches; water cylinder, 6 inches; stroke, 12 inches; capacity per stroke, 1.469 gallons.

Other auxiliary machinery.—In each engine room, there is one horizontal, double, double-acting, yoke, fly-wheel pump, $8\frac{1}{2}$ by $6\frac{1}{4}$ by 10 inches, for fire and bilge service.

In each engine room, one horizontal, single, double-acting, Dow pump, 10 by 6 by 12-inches for auxiliary purposes. These pumps draw from either the sea or the hotwell, and deliver into the water-service pipes, into the fire main, or into the feed tanks.

In each engine room, one double vertical engine, 5 by 5 inches, for turning the main engines.

In each engine room, one ventilating fan and engine. The engine is simple, direct-acting, horizontal. Steam cylinder, 4 by 4-inches, diameter of fan, 3 feet 9 inches.

In the port engine room, one three-cylinder, vertical, direct-acting, plunger-pump for steering gear, 8 by 2¼ by 8 inches. The steam cylinders are double-acting; capacity 41 gallons per 100 revolutions.

In the starboard engine room, one beam, fly-wheel air-compressor. The steam cylinder is vertical, 5 by 12 inches, directly connected to one end of the beam. The other end of the beam actuates the air compressors; diameters, 1½ and 4-inches by 12 inches stroke.

This air compressor serves as a pressure regulator for the hydraulic pumping engines and the steering engine, acting in a manner to be described below.

For each boiler room, one double ash-hoist, 4 by 6 inches. These engines are located in the distiller room, which is in the superstructure, above the armore l deck.

In the starboard fire room, is one horizontal No. 00 Blake pump for supplying water to the firemen's wash water tanks.

On the berth deck, just forward of the berth compartments, are two Sturtevant blowers for ventilating the ship. Diameter of fan 4 feet 6 inches, driven by two simple, inclined engines, 6 inches diameter of cylinder by 6 inches stroke.

On the berth deck, forward, a 10 by 18-inch two-cylinder reversible windlass with two wildcats.

On the superstructure deck, one winch for handling boats. Steam cylinder 8 by 8 inches.

In the Engineers' workshop, one 4-inch by 5-inch engine for driving the workshop machinery.

Distilling apparatus.—There is one Baird low-pressure evaporator, of the horizontal type, and a Baird four-coil distiller with filter.

One combined air and circulating pump for circulating water through the distiller, creating a vacuum in the distiller coils and in the evaporator, and delivering the fresh water into the filter. Steam cylinder, 6½-inches; air, 10-inches; water, 7-inches; stroke, common, 10 inches.

One combined brine and fresh water pump. Size of cylinders, steam, 3 inches; fresh water, 2 inches; brine, $1\frac{3}{4}$ inches; stroke, 3 inches. The fresh water side of this pump takes fresh water from the filter, draws it through a water meter, and delivers into the fresh water tanks. The brine pump removes the highly saturated water from the evaporators.

Feed water for the evaporators is drawn from the water that has circulated through the distiller. A by-pass is provided so that, when distilling for the main boilers, the water can be passed directly to the feed tanks without going through the filter. There is, besides, a direct connection from the evaporator to the auxiliary exhaust pipe, and a branch from the auxiliary steam pipe to the distiller.

Feed tanks.—There is in each boiler room a large feed tank, capacity about 1200 gallons, to act as a storage reservoir of fresh water for the boilers. These tanks are provided with two perforated diaphragm plates near their top, between which is placed filtering material consisting of bags of hay wrapped in blankets. The auxiliary drain pipe, conveying the water drained from auxiliary engines and traps, discharges either into the condenser, the bilge, or the feed tanks. At the discharge into the latter, Macomb strainers are provided containing filtering material for straining out grease and other impurities.

Electric Light Plant.—There are three Edison compound multi-polar (eight poles; four inside, four outside) marine dynamos of 200 ampères and 80 volts, at 400 revolutions. There are 384 incandescent lamps (16 c. p.), of 6,144 total candle power, arranged on twelve circuits. There are four additional circuits for as many Mangin projectors. These were made by Sautter, Lemonnier et Cie, on four mounts. Each dynamo is solidly coupled to an inverted, inclined, compound engine, with cylinders $7\frac{1}{2}$ and 12 by 6 inches.

Hydraulic Machinery.—Hydraulic power is used for turning the turrets, elevating the guns, checking the recoil of guns, and running them out, loading and hoisting the ammunition. Hydraulic power is also employed for operating the rudder and the battle hatches. For these latter purposes power is ordinarily

furnished by the three-cylinder pump already described. The pumping engines for moving the turrets and working the guns may, however, be used for these purposes also.

There are two independent sets of pumping engines—one for the forward, and one for the after turret; the pressure pipes from the two sets are, however, connected, so that either set of pumps can be used on either or both turrets. Each set of pumping engines consits of two independent piston pumps, built by the Dow Pump Works, of San Francisco, and provided with the Dow valves and gear. The forward pumps are 26 by 10 by 27 inches; the after pumps 24 by 8½ by 20 inches. The pumps are horizontal, double acting, the pump rod continued into a tail rod. The forward pumps have four receiving and four delivery valves at each end, the after pumps have three. These valves are of rubber in phosphor-bronze casings. The working pressure for turret as well as for the steering engine pumps is 600 pounds, and the regulation is effected by means of the air compressors previously described. The air compressor compresses the air to 600 pounds, or whatever water pressure is to be employed. Pressure pipes lead from the compressor to regulating cylinders at the pumps. For the turret pumps, these consist of a casing 17 inches diameter inside by about 3 feet long. Within this is a cylinder of 6 inches bore and 2 feet 5 inches long, in which works a piston having the constant air pressure on the lower side and water pressure on the upper by a pipe connecting this space with the pressure main. The piston rod, by a suitable mechanism, actuates the stem of a throttle valve, and regulates the supply of steam to the steam cylinders of the pumps.

The engines for turning the turrets, two for each turret, are single-acting, three-cylinder engines, cylinders at 120 degrees, actuating a common crank pin. By means of intermediate gearing, a pinion is driven, meshing into a stationary circular rack. The distributing valves are slide valves. The operating gear consists of a hand wheel and screwed spindle actuating one end of a beam, the spindle of the reversing valve being jointed to the center, while the other end of the beam receives opposite motion from a second screw spindle actuated by a pinion meshing into

the stationary rack. The turning engines are 7¼ by 10 inches for the forward and 7¼ by 7½ inches for the after turret.

Coal Bunkers.—The capacities of the various compartments of the coal bunkers are as follows:

		Starboard. cubic feet.	*Port. cubic feet.*
Athwartship bunker, No. 1		814	814
Fore-and-aft bunker	No. 2	962	962
	No. 3	937	937
	No. 4	1,469	1,304

Total capacity of all bunkers, 8,199 cubic feet, or 195.2 tons of 42 cubic feet.

WEIGHTS.

Propelling machinery	356.24 tons.
Water in above machinery	47.16 tons.
Tools, instruments and duplicate parts	18.91 tons.
Miscellaneous machinery (includes all machinery under this head on the ship, except the hydraulic machinery)	52.37 tons.
Hydraulic machinery (includes pumping engines and connections, air compressor and turret-turning engine and gear)	40.86 tons.
Total weight at time of completion	515.54 tons.
One cylindrical boiler with fittings, empty	24.07 tons.
Water in one cylindrical boiler	12.80 tons.
One Ward boiler with fittings, empty	15.09 tons.
Water in one Ward boiler (2 gauges)	2.35 tons.
All boilers, with fittings and water	143.50 tons.
All boilers, fittings, uptakes, smokepipes, and all machinery in fire rooms	211.34 tons.
Reciprocating parts:	
H.P. piston	516.62 pounds.
I.P. piston	1,171.00 pounds.
L.P. piston	2,241.75 pounds.
Piston rods	431.50 pounds.
Crossheads	789.75 pounds.
Connecting rods	1,654.37 pounds.

TRIAL.

The trial took place on January 5th, between 9.46 A. M. and 1.46 P. M. The course run was between Hunter's Point and Red Rock in San Francisco Bay. The length of the course was such that it would take a little more than half an hour to run over it. Cards were taken when the ship went on the course, again at the end of ten minutes, and again at the end of the second ten minutes. No cards used in the computation of results were taken while the ship was making the turns at each end of the course. During the four hours of the trial there was no change in the

adjustment of the throttle or cut-off. The engines worked well during the entire run, and no water, except the regular circulation through hollow brasses and guides, was used on any of the main or auxiliary machinery, excepting on engines of fire-room blowers. The boilers steamed freely and furnished all the steam required during the first part of the run, but owing, in a great measure, to the unskillful handling of the fires, the steam pressure fell off towards the latter part of the run. The reducing valves for equalizing the pressure between the Ward and cylindrical boilers were secured off their seats, rendering them inoperative. The Ward and cylindrical boilers were consequently worked in battery in the same manner as if all the boilers had been cylindrical. The Ward boilers carried their water steadily and generated steam regularly, and without any sudden fluctuations of pressure.

The boilers and engines were examined after the trial, and were found to be in excellent condition with the exception of the starboard cylindrical boiler, which showed leaks around back tube sheet, and slight bulging of back sheet of combustion chambers.

The contractor's force in the Engineer's department consisted of the following:

One superintending engineer, in charge of all the machinery in the ship.

Four engineers holding chiefs' certificates.

Six machinists.

Twenty-two oilers, water tenders, engine tenders, storekeepers, &c.

Twenty-two firemen and coal passers.

The turning trials showed that a half-circle was turned in two minutes and fifty seconds, and a complete circle in five minutes and thirty seconds. Ten seconds were required to shift the helm from amidships to hard over by hydraulic gear when the ship was at full speed, and 51 seconds by hand under the same conditions.

It required ten seconds to shift from hydraulic to hand-steering gear.

An attempt was made to determine the amount of coal burnt during the trial, but the results obtained are not accurate enough to warrant publication.

The coal used was Harris' Navigation, Welsh, semi-bituminous, of the best quality.

DATA OF TRIAL.

Draught at beginning	forward, 13 feet 5 inches; aft 15 feet 6 inches ... 14 feet 5½ inches.
Draught at end	forward, 13 feet 4 inches; aft 15 feet 5 inches ... 14 feet 4½ inches.
Draught mean for trial	forward, 13 feet 4½ inches; aft 15 feet 5½ inches ... 14 feet 5 inches.
Displacement at mean draught on trial	4,000 tons.
Area of immersed midship section	772 square feet.
Wetted surface	19,203 square feet.
I.H.P. (total) per 100 square feet of wetted surface	27.31
I.H.P. per 100 square feet of wetted surface at 10 knots in ratio of 3.5 power	10.73
Average speed	13.6 knots.
Slip (mean of both screws)	18 73 per cent.
Speed^3 × area immersed midship section ÷ I.H.P.	370.3
Speed^3 × displacement$^{\frac{2}{3}}$ ÷ I.H.P.	120.9

SYNOPSIS OF STEAM LOG.

		Starboard.	*Port.*
Revolutions of main engine per minute		162.86	161.17
Revolutions, mean of both engines		161.51	
Piston speed, feet per minute		814 3	805.85
Steam at boilers (gauge)		155	
Steam at engines (gauge)		153.05	148.6
Steam at first receiver (absolute)		72.2	68.3
Steam at second receiver (absolute)		29 7	29.5
Vacuum in condenser in inches of mercury		26.6	26 23
Opening of throttle		Wide.	
Steam cut off in fraction of stroke from beginning,	H P.	.74	.74
	I.P.	.74	.74
	L.P.	.72	.72
Double strokes of air pump per minute		85.6	84.72
Revolutions of circulating pump per minute		327.	344.
Temperatures in degrees Fahrenheit,	Steam at H.P. cylinder	359.3	337.8
	Engine rooms	109.3	109.4
	Injection	49.7	
	Discharge	81.	78.9
	Hot well	98.5	96.0
	Feed	115.1	139.2
	Fire rooms	145.5	131.1

		Starboard.	*Port.*
Revolutions of blowers per minute		558.0	682.
Air pressure in fire rooms in inches of water		3.16	3.28
H.P. cylinder,	Mean pressure	52 55	54.96
	I.H.P	734.43	760.18
I.P. cylinder,	Mean pressure	23.32	22.73
	I.H.P	748.21	731.22
L.P. cylinder,	Mean pressure	12.85	12.61
	I.H.P	1,021.26	991.56
Aggregate equivalent mean pressure on L.P. piston		31.53	31.60
Collective I.H.P. each main engine		2,503.90	2,482.96
Collective I.H.P both main engines		4,986.87	
I.H.P. of air pumps		18.37	18 16
I.H.P. of circulating pumps		22.37	26.00
Collective I.H.P. of air and circulating pumps		84 90	
I.H.P. main blowers		83.91	
I.H.P. main feed pumps		32.28	
I.H.P other auxiliaries		55.96	
Collective I.H.P. all auxiliaries		257.06	
Collective I.H.P. each main engine, air and circulating pumps		2.544.64	2,527.12
Collective I.H.P., both main engines, air and circulating pumps		5,071.77	
Total I.H.P. main engines and all auxiliary machinery in operation		5,243.92	
Indicated thrust (I.H.P. main engines)		43.490 5	43,579.
Indicated thrust per square foot of developed area (39 square feet) of propeller		1,115.1	1,117.4
Indicated thrust per square inch of surface of thrust bearing		59.9	60.05
Cubic feet swept per minute by L.P. piston per I.H.P.		7.26	7.25
Square feet of cooling surface per I.H.P.		1.46	
Square feet of total heating surface per I.H.P.		2.81	
I.H.P. all machinery in operation per square foot of G.S.		13.68	
I.H.P. per ton of boilers, water, fittings, and all fire room weights		24.7	
I.H.P. per ton of propelling machinery, boilers and water		12.95	
Velocity of steam, in feet per second, through the ports at the maximum opening of the valves:			
H.P. steam port	Top	118.8	117.56
	Bottom	152.06	150.4
I.P. steam port	Top	136.76	135.3
	Bottom	175.6	156.5
L.P. steam port	Top	209.2	207.0
	Bottom	215.7	213.2
H.P. exhaust port		95.03	94.1
I.P. exhaust port		109.1	108.

	Starboard.	*Port.*
L.P. exhaust port	137.6	136.27
Main steam pipe	137.7	136.27
Main exhaust pipe	138.24	137.9

MAXIMUM PERFORMANCE.—CARDS NO. 6.

Revolutions of main engines per minute	167.6		165.4
Revolutions, mean, of both engines		166.5	
Piston speed, feet per minute		832.5	
Steam at boilers, gauge		162.0	
Steam at engines, gauge	160.0		160.0
Steam at first receiver, absolute	77.		74.
Steam at second receiver, absolute	32.		32.5
Vacuum in condenser in inches of mercury	26.25		26.3
Opening of throttle	wide		wide
Steam cut-off in fraction of stroke from beginning H.P.	.74		.74
Steam cut-off in fraction of stroke from beginning I.P.	.74		.74
Steam cut-off in fraction of stroke from beginning L.P.	.72		.72
Double strokes of air pump per minute	87.8		78.5
Revolutions of circulating pump per minute	336.		353.
Temperatures in degrees Fahrenheit, Steam at H.P. cylinder	344.		338.
Temperatures in degrees Fahrenheit, Engine room	106.		106.
Temperatures in degrees Fahrenheit, Injection	51.		51.
Temperatures in degrees Fahrenheit, Discharge	80.		80.
Temperatures in degrees Fahrenheit, Hot well	98.		96.
Temperatures in degrees Fahrenheit, Feed	118.		140.
Temperatures in degrees Fahrenheit, Fire room	140		132.
Revolutions of blowers per minute	561.5		665
Air pressure in fire room in inches of water	3 2		3 2
H.P. cylinder, Mean pressure	56.35		58.31
H.P. cylinder, I.H.P.	809.41		827.21
I.P. cylinder, Mean pressure	24 43		24 02
I.P. cylinder, I.H.P.	816.32		792.40
L.P. cylinder, Mean pressure	13.7		13.69
L.P. cylinder, I.H.P.	1,119.45		1,103.83
Aggregate equivalent mean pressure on L.P. piston	33.55		33.77
Collective I.H.P. each main engine	2,745.18		2,723.45
Collective I.H.P. both main engines		5,468.63	
I.H.P. of air and circulating pumps	40,74		44.16
Collective I.H.P. of air and circulating pumps		84.90	
I.H.P. of all other auxiliaries		172.15	
Collective I.H.P. all auxiliaries		257.06	
Collective I.H.P. each main engine, air and circulating pumps	2,785.93		2,767.61
Collective I.H.P. both main engines, air and circulating pumps		5,553 54	

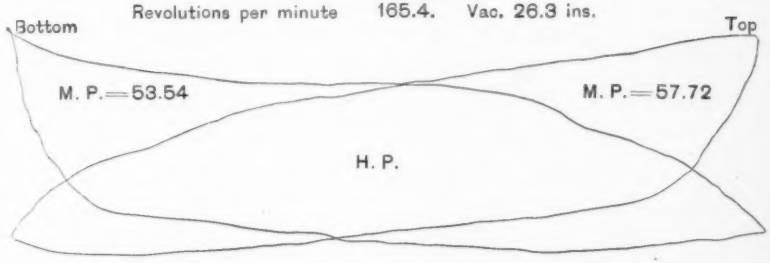
U.S.S. MONTEREY.

PORT ENGINE.

Jan. 5th, 1893. 10.50 A. M.

Steam pressure at engines 160 lbs.

Revolutions per minute 165.4. Vac. 26.3 ins.



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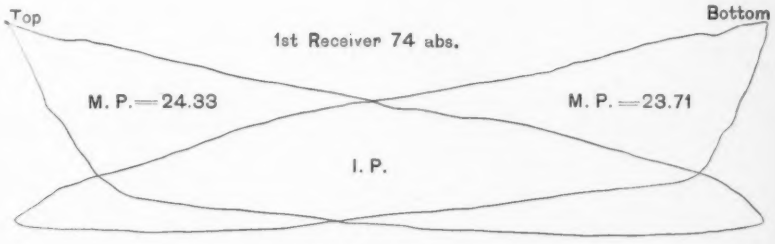

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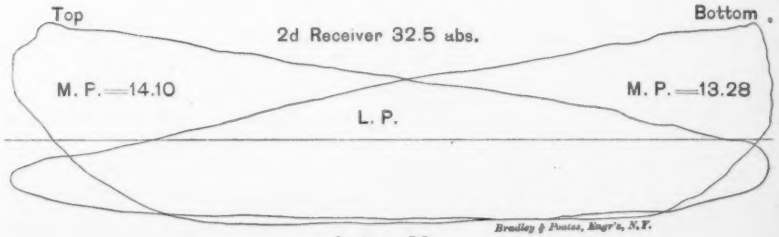

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To
Bott

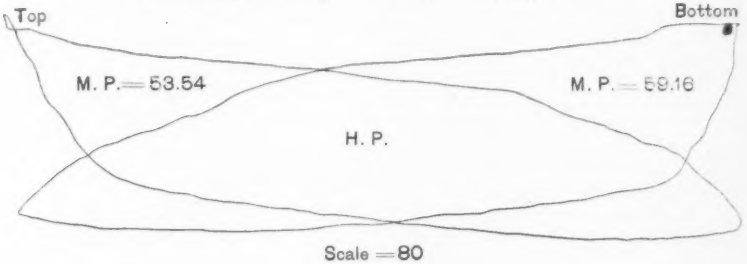
U. S. S. MONTEREY.

STARBOARD ENGINE.

Jan. 5th, 1893. 10.50 A. M.

Steam pressure at engines 160 lbs.

Revolutions per minute 167.6. Vac. 26¼ ins.



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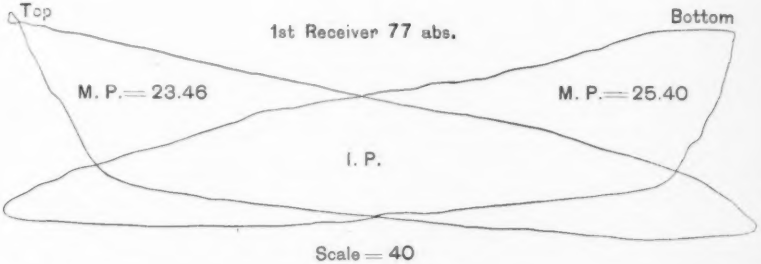

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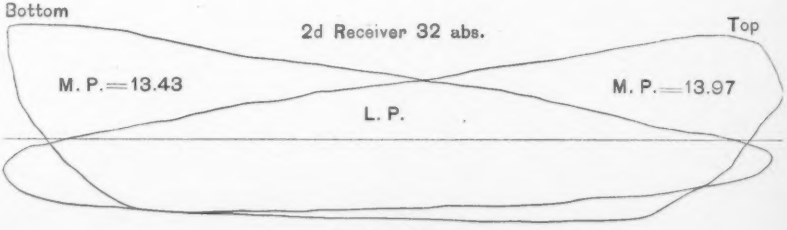


Scale = 20

	Starboard.	*Port.*
Total I.H.P. main engines and all auxiliary machinery in operation	5,810.60	
Cubic feet swept per minute by L.P. piston per I.H.P.	6.8	6.6
Square feet of total H.S. per total I.H.P.	2.54	
I.H.P. all machinery in operation per square foot G.S.	15.16	
I.H.P. per ton of boilers, fittings, water and all fire-room weights	27.38	
I.H.P. per ton of propelling machinery, boilers and water	14.34	

NOTE.—The indicated horse power used in calculation of horse power per ton of machinery is the total power developed by all steam machinery in use during the trial, less that developed by the steering engine, distiller pump, dynamo engines and air compressor. The weight of these engines is not included in weight factor of the calculation.

The cards shown are for the maximum performance of the main engines.

Our thanks are due to Mr. Leo Morgan, draughtsman for the Government at the Union Iron Works, for the preparation of much of the data given.

THE GREAT LAKES OF NORTH AMERICA.

By George H. Ely, C. E.

[Extracts from a paper read at the International Congress on Inland Navigation, Paris, 1892.]

For about one-half the distance across the American continent the waters of the Saint Lawrence system divide the Dominion from the Great Republic. The boundary line, beginning on the St. Lawrence in latitude 45°, passes through the middle of Lakes Ontario, Erie, St. Clair, Huron, the St. Mary's River and Lake Superior to a point on its north shore, 124 miles east of Duluth and Superior, the western end of Lake Superior. Lake Michigan is wholly within the territory of the United States.

These great American lakes contain more than one-half the area of all the fresh water of the globe. They make up the largest system of deep-water inland navigation on the globe. The water area is 95,460 square miles; the drainage area is 150,000 square miles. No other inland water-way bears upon its bosom so vast a commerce or touches, as this does, the vital interests of so many millions of men. The topographical relations of these connected waters are very remarkable. Lying, in general direction, east and west, between the 41st and 47th parallels, they penetrate from tide water on the St. Lawrence and (including the Erie canal) from tide water at New York 1,400 miles into the heart of the continent. The head of Lake Superior and the St Lawrence tide water are on the northernmost parallel; Chicago and New York on the southern. The western extremity of the system, the head of Lake Superior, is 1,700 miles only from the waters of the Pacific. It is 2,384 miles from Belle Isle, at the mouth of the St. Lawrence, and 4,618 miles from Liverpool.

The range of this water system, it will be observed, is entirely within the limits of the north temperate zone, on the line on

which population has most freely moved westward, where final settlement is most compact, and where climatic conditions insure the largest returns to capital and labor. Lake Superior, the head of the system, alone receives the waters of 200 rivers, and drains 85,000 square miles.

One hundred and fifty miles northwest of Duluth and Superior are the fountains of three of the great drainage systems of the continent. Physical conditions, *there*, send waters flowing northward to the ocean, through Hudson's Bay; southward, through the Mississippi Valley and the Gulf of Mexico; and eastward, through the Lakes and the St. Lawrence. For commercial purposes, the northern drainage system has not yet been utilized; but flowing water will forever be a potent instrument of commerce, southward and eastward, between the interior and the Atlantic coast.

Such are the peculiar and favoring physical conditions under which these two great peoples of the English tongue occupy, side by side, the North American continent from ocean to ocean, using in common this continental water-way, and, by treaty stipulations, interchanging with each other the use of improvements inside their respective boundary lines.

The tonnage now moving on these lakes originates, in very large proportion, on the soil of the United States. This is due partly to advantages in respect to climate, proportion of arable land, and population, which are heavily on the side of the Republic. Eight of the States of the Union, some of them our largest, most populous and most highly productive, border directly upon them, while beyond the head of Lake Superior, tributary to it, are six hundred thousand square miles more of American territory. North of this is another like empire, under the British flag, fair but yet untouched—1,000 miles square, and capable of boundless production of cereals—tributary also to these waters.

From both sides, then, of this continental boundary line, inevitably and forever, will come here, for transit into the world's commerce, the products of the vast plains and the mountain region of the far Northwest.

On this line also, to a large extent, will be made the commercial exchanges of the Pacific slope, Australia, China and Japan, with the Atlantic States and with Europe.

The difference in level between Lake Superior and tide water on the St. Lawrence is about 600 feet. This has been overcome by several locks and canals. For this purpose, and for deepening of channels and the removal of obstructions, heavy expenditures have been made by both nations—each inside the limit of its own territory. On the American side are the canal and lock at Sault Ste. Marie and the St. Clair Flats canal, the latter being an excavation only through shallow portions of Lake St. Clair, the banks being protected by piling and without locks.

On the Canadian is the Welland, connecting Lake Erie and Lake Ontario, 26¾ miles in length, and having 14 feet of water on the sills of locks; and on the St. Lawrence, above Montreal, there are six short canals, aggregating in length 43⅝ miles. The depth of water in these, now available for navigation, is less than 10 feet.

The enlargement and deepening of these canals to the capacity of the Welland is now in progress, and will be completed within four years.

By the completion of the ship channel through Lake St. Peter, Montreal has now 27½ feet of water to the ocean.

The Dominion government has also begun the construction of a lock and canal, on the Canadian side, at Sault Ste. Marie. This lock is to be 900 feet in length, 60 feet wide, and with 21 feet water on the miter sill.

Lake Superior was unknown, unexplored, long after population had moved into the territories bordering upon the other lakes of the system. It lay in distant isolation, enfolded by a wilderness, the coming civilization heralded only by the missionary and the fur trader coasting along its silent shores. Its mineral treasures first drew the explorer up the St. Mary's River. How faint, then, the conception of the commerce destined to pass through that channel! The descent in the St. Mary's River, 75 miles long, is 20 feet 4 inches; 18 feet 2 inches of this is at the Sault. Two feet 2 inches are distributed over the first 35

miles below the Sault. Lake Superior was opened by the completion of the canal and first lock in 1855. This passed vessels drawing a maximum of 11½ feet water. This was superseded in 1881 by the lock now in use—515 feet long, 80 feet wide, and 17 feet water on the miter sill. From that time this lock has, with wonderful efficiency, met the demands of a rapidly-increasing tonnage. In 1882, it passed 2,029,000 tons; in 1883, 2,267,000; in 1884, 2,874,000; in 1885, 3,256,000; in 1886, 4,527,000; in 1887, 5,494,000; in 1888, 6,932,000; in 1889, 7,516,000; in 1890, 9,041,213 tons, [in 1892, 11,241,000 tons] the average season of navigation in that latitude being about 220 days. The total number of passages in 1890 was 10,557. The average number passing per day was 45.3. The average registered tonnage per vessel in 1887 was 626.3; in 1891, it was 862.1. The annual average net tonnage for the last five years of the Suez Canal—a world's channel of commerce, and open every day in the year—was 6,983,166 tons. The annual average net tonnage of the lock and canal at Sault Ste. Marie for the same period—open only an average of 220 days in the year—was 6,821,062 tons.

In 1886 appropriations were begun for the third lock, to be placed alongside the lock now in use, and on the site of the original lock of 1855. This work, now under construction by General O. M. Poe, and to be completed in 1896, at a cost of $4,988,865, is 800 feet long, 100 feet wide and 21 feet on the sill. The completion at an earlier date of the Hay Lake channel, a supplementary work in the St. Mary's River, below the lock, at a cost of $2,659,000, insures 20 feet of navigable water through the St. Mary's River to Lake Huron, and a saving in distance of 11 miles.

Twenty feet of water here, however, means, and was always intended to mean, 20 feet through every other channel on the lakes between Chicago and Duluth and Buffalo.

In pursuance of this comprehensive plan, Congress has begun appropriations for the removal of obstructions at six different localities along the line. The estimates for this work are $3,394,835, and 20 feet of water will be secured at each of these locali-

ties contemporaneously with the completion of the works at the Sault, in 1896. The localities and the estimates are as follows:

Removing obstructions at Sailors' Encampment	$556,333 00
Dredging at St. Clair Flats	313,559 40
Dredging at Grosse Point Flats	956,825 76
Dredging at mouth of Detroit River	977,850 00
Removing obstructions at foot of Lake Huron	449,512 80
Removing two shoals near Round Island	140,755 00
	$3,394,835 96

The registered American tonnage of the lakes, June 30, 1891, was 1,154,870 tons, 1,592 steam vessels, representing 736,751 tons, and 2,008 sail, 418,118 tons. The tonnage has more than doubled in the last five years, the increase being almost exclusively in steel steamships of 1,500 to 2,500 tons register. The shipbuilding plant of the port of Cleveland alone launched 71,322 registered tons in the years 1889–1890. As a shipbuilding locality, Cleveland has already become second only to the Clyde.

During the financial year ending June 30, 1892, 64 vessels, representing 83,223 net registered tons, mainly steel steamships, have been added to the Lake fleet, at a cost of $1,911,000. Of the total U. S. tonnage launched in 1889, 46 per cent. was on the Great Lakes, 41 per cent. on the Atlantic Coast, 8 per cent. on the Pacific Coast, 5 per cent. on the Western rivers. The capacity and power of many of these vessels have been indicated by General Poe, as follows: "The increase in the available depth of channels on the lakes, from nine and one-half feet in 1852 to sixteen feet in 1882, developed this commerce, and it is only reasonable to expect that a further increase of four feet will be followed by a corresponding increase in the shipping. The results are more notable, perhaps, in the character of the vessels employed in the carrying trade. These have increased in size and seaworthiness until they form a fleet which has not its equal upon any inland waters on the face of the globe.

"Of large capacity and great power, regardless of wind or weather, the steamers of the prevailing type bear their cargoes to and from ports, a thousand miles apart, with the precision of

railroad trains, each of them transporting at once more than ten ordinary freight trains."

The capacity and power of some of our latest constructions are illustrated by the sister steel steamships *Maritana* and *Mariposa*, just put afloat for the Minnesota Steamship Company. The former was launched by the Chicago Shipbuilding Company, June 8, 1892, and the latter by the Globe Iron Works Company, Cleveland, June 16. The dimensions of these boats are: Length of keel, 330 feet; length over all, 350 feet; breadth of beam, 45 feet; depth of hold, 24½ feet. The engines are triple-expansions, 24, 39 and 63 inches, with 48 inches stroke, steam being furnished from three steel boilers 12 feet in diameter by 12½ feet long.

These ships are guaranteed to carry 3,000 gross tons on the present draught of water, 14½ feet, and 4,000 gross tons on 17 feet. The twenty feet of water, expected in 1896, will, of course, correspondingly increase the carrying capacity of these ships.

An interesting statement of the dimensions and construction of the Minnesota Steamship Company's steel steamer *Manola* and of her work in the season of 1890, in all its details, will be given in the appendix.

The work of this lake shipping is given *approximately* by the United States census report of 1890. The freight movement in 1889, on all the lakes was estimated by that report at 53,424,432 tons. The tonnage put afloat since then has increased this movement to 63,240,514 tons. *Estimates only* can be given, because at one point only on the lakes, at Sault Ste. Marie, is there an official record made of tonnage movement. The movement through the Detroit river alone, in 1889, was estimated at 36,203,586 tons. The grain movement alone, in the season of 1891, from the Lake Superior ports, Duluth and West Superior, and from Chicago, Milwaukee, Toledo and Detroit, was 197,165,208 bushels. The average lake freights for the last three years on wheat from Chicago to Buffalo has been 1.73 cents per bushel. From Chicago to New York, by the lakes and the Erie canal, 6.87 cents. The all-rail average rate from Chicago to New York the last three years has been 14.76 cents per bushel.

The total entries and clearances, foreign and coastwise for the port of London that year (1889) were 19,245,417 tons; of Liverpool, 14,175,200 tons. The total entries and clearances in *the foreign trade* at New York that year were 11,051,236 tons; in the foreign trade, at all the seaports in the United States, 26,983,313 tons.

The estimate of the tonnage movements through the Detroit river in 1889, then, was at least 10,000,000 tons above the total entries and clearances in the foreign trade at all the seaports of the United States, and three million tons above the combined foreign and coastwise tonnage of the ports of London and Liverpool.

The ton mileage of the lake traffic in 1891 was more than 25 per cent. of the total ton mileage of all the railways of the United States.

The value of this American shipping now afloat on these waters is $68,925,449.

This extraordinary growth in shipbuilding and commerce on the lakes implies corresponding changes of conditions as to population and production along the thousands of miles of their shore line and in the tributary country.

Such equipment and use of these waters mean industrial activity and large advance in population.

	1880.	1890.
Four cities on Lake Superior had population	5,528	64,147
Five cities on Lake Huron and Lake Saint Clair	181,610	304,863
Twelve cities on Lake Michigan	734,196	1,502,663
Seven cities on Lake Erie	420,685	675,310
	1,342,019	2,549,683

An increase of population in ten years of 85 per cent.

Of the total tonnage moving through the Sault canal, 90 per cent. is made up of five commodities—wheat, corn, iron ore, coal and lumber. These, it will be observed, are all *primary products*, whose bulk and weight are large in comparison with value. They went to market in 1891—over one hundred million dollars in value—on the average draught of water that year at the Sault lock of 14 feet 8 inches, an average distance of 800 miles, at a cost of one mill and three-tenths per ton per mile.

This is less than *one-fifth of the lowest cost* reported by any railroad in the United States. The average cost of transportation on all the railways of the United States in 1890 was over nine mills per ton per mile. The difference between this cost of transportation on the tonnage moved that year on the lakes and the water rate, as given above—distributed in some proportion between the producer and the consumer—would have been $147,027,514. But, as to a large portion of this tonnage, any possible cost on wheels would not have permitted it to move at all. In such a case, its production at the point of origin would, of course, have been impossible. That, in turn, would have halted the pioneer emigrant *this side* the richest areas of the continent.

Of this, Lake Superior iron ore furnishes a conspicuous example. General Poe reports that "for the whole period since 1881" (when the present lock carrying about fifteen feet navigable water was opened), "the iron ore carried through the canal has been 47 per cent. of the total freight, and in 1889 and 1890 it was more than 50 per cent." But Lake Superior iron ore could never have gone on wheels to meet the coal of the central States.

Deep-water communication, then, with the Northwest *alone* rendered the present development of Lake Superior iron ore possible. These ores have been a chief factor, under our protective policy of the last thirty years, in the enormous development of our American iron and steel industries.

We are now the largest producers of pig iron, Bessemer steel ingots and Bessemer steel rails in the world.

We consumed in the United States, in 1890, 17,500,000 tons of iron ore. Of this, 1,246,830 tons were imported; 16,253,170 tons were of home production. Of the home production, the Lake Superior region furnished 9,003,701 tons.

Out of Northwest, then, over these lakes, come more than one-half the ores for the vast iron and steel industries of our country. They are distributed mainly through Ohio, Pennsylvania, West Virginia, New York and Illinois.

The four mining districts, all lying on and accessible from Lake Superior and Lake Michigan (the Marquette in Michigan, the Menominee in Michigan and in Wisconsin, the Goge-

bic in Wisconsin and Michigan, and the Minnesota) produce a great variety of ores which are known in the market as "hard" and "soft," and "Bessemer" and "non-Bessemer;" but all are noted for richness in iron, and for the absence of substances deleterious to iron and steel. The very high grade of these ores, and the low cost at which deep-water navigation brings them to the coal accessible from the lower lakes, explains the enormous demand for them and their position in the iron and steel trade of the United States.

Their chemical constitution will be indicated by the following table of analyses of four hundred cargoes (being about 800-000 tons) delivered at lower lake ports, not including ores from the new developments on the Messaba range.

"*Average analyses of Lake Superior Ores, No. 1.* Best grade, comprising 85 per cent. of the total hard ore production, and 90 per cent. of the total soft ore production. The analyses are averages of 100 cargoes of each kind of ore mentioned, and the samplings for same were made by Rattle & Nye, of Cleveland, Ohio, during the shipping season of 1889, at the principal lower lake receiving ports, viz., Cleveland, Ashtabula, Fairport, Erie, Lorain, Buffalo, Sandusky and Toledo."

Designations.	Metallic Iron.	Silica.	Phosphorus.	Alumina.	Lime.
Hard Bessemer	67.55	2.05	0.041	0 35	0.080
Hard Non-Bessemer	67.22	2.51	0.147	0.27	0.122
Soft Bessemer	62.68	5.90	0.040	2.09	0.224
Soft Non Bessemer	62.71	6.12	0.215	1.99	0.186

Designations.	Magnesia.	Sulphur.	Manganese.	Organic and Volatile Matter.
Hard Bessemer	0.111	0.028	0.124	0 48
Hard Non-Bessemer	0.141	0.033	0.142	0.52
Soft Bessemer	0.142	0.030	0 440	1.78
Soft Non-Bessemer	0.213	0.048	0 371	1.42

About 100 mining locations are now in operation in the Lake Superior districts. The Marquette district was opened in 1855; the Menominee in 1878; the Gogebic and the Minnesota in 1884.

The total production since 1855 (1891 inclusive) has been 65,462,737 gross tons.

The magnitude of the investments in Lake Superior ore mining and in its distribution will appear from the following:

Capital in the mines in the four Lake Superior districts in 1889, as per U. S. Census	$54,825,122
Increase to July 1, 1892	15,000,000
	$69,825,122
Capital in docks and their equipments at Lake Superior and Lake Michigan ports, built and used exclusively for shipping ore (official)	9,885,665
Capital employed exclusively in railroad transportation of ore from mines to shipping ports on Lake Superior and Lake Michigan (official)	27,014,594
Floating capital on the Lakes, employed exclusively in ore transportation from Upper Lake to Lower Lake ports (estimated)	29,933,107
Capital in docks and in their equipments for receiving and forwarding ore exclusively at Lake Erie ports, between Toledo and Buffalo, exclusive of both (official)	12,392,880
Capital employed exclusively in railroad transportation of ore, inland to mills and furnaces, from Lake Erie ports (official)	26,343,617
	$175,394,985

The far-reaching relations of this mass of capital along the whole range of interdependent industries between the mines and the furnaces and mills cannot be dwelt upon within the limitations of this occasion, but an allusion to them is both natural and fitting, because they are the creation of a system of a wise improvement of internal waterways.

There are two great questions respecting this waterway of profoundest interest to the American people. A brief reference to them should be made in closing this paper. One relates to maintaining these waterways *unobstructed;* the other to their enlargement and extension.

Strange as it may seem, during several years past successive attempts have been made in the interest of railroads converging to the Detroit river, to secure from Congress bridge franchises at that point of the concentrated traffic of the entire water system. To the Detroit river, there is no alternative channel; through that passage, 2,600 feet wide only—and a much less width actually available for deep-draught navigation—sweeps the floating tonnage of a continent.

The alarm and strenuous resistance of the people have hitherto kept this channel free of all obstructions. Congress has three times—in 1873, in 1879, and in 1890—rejected all bridging propositions.

The producing and navigation interests of the country insist upon the entire feasibility of tunnels *beneath* this channel. In this position they are absolutely sustained by the fact that the traffic of the Grand Trunk Railroad Company is now passing freely under these waters, sixty-five miles above, at Sarnia and Port Huron.

These interests have hitherto successfully defended this open waterway, and it may now be assumed that its usefulness will never be impaired by such *direct* or *contingent* obstructions.

The other question is the extension of deep-water navigation eastward, from the foot of Lake Erie, by way of the Hudson River, to the ocean at New York. The logical sequence of 20 feet of water through the lakes west of Ontario is an outlet, of equal depth, eastward to the Atlantic.

The St. Lawrence route involves a shorter season of navigation.

It terminates at no center of commercial exchanges.

Its improvements now in progress contemplate fourteen feet of water only.

It is obvious that this will never meet the demands of American commerce.

Again, while there is no reason to apprehend any disturbance of the existing friendly relations between Canada and the United States, *dependence* in commercial relations is not the best guaranty of independence in political relations.

The deep-water extension to the ocean, then, must be through American territory. The agency of the Erie canal, with its successive enlargements during the last sixty years, in reducing the cost of transportation, is incalculable; but a ship channel to the Hudson, either on the line of the Erie canal or by way of Niagara Falls and Lake Ontario, is now required. The Detroit Deep Waterways Convention, of December, 1891, appealed successfully to Congress for a survey of routes, and an appropriation for this purpose has been reported.

The cost of this work, by whatever route, will be immense, but its relations are so vital that the expenditure *must* and *will* be made. Then, without breaking bulk, our lake shipping will pass out over all oceans, and share the world's commerce with the strongest.

Appendix.

MINNESOTA STEAMSHIP COMPANY'S STEEL STEAMER *MANOLA*.

Length, keel, 292 feet 5½ inches; length over all, 308 feet 5½ inches; breadth, 40 feet; depth (molded) 24 feet 6 inches; size, of cylinders, 24, 38 and 61 inches, and stroke 42 inches. Two boilers of 14 feet 0 inches diameter, 12 feet 6 inches long; one wheel of 14 feet 0 inches diameter, 16 feet 6 inches pitch; size, water bottom, 3 feet 4 inches deep, divided into 6 compartments, independent of collision bulkhead; capacity water bottom, 800 tons; estimated horse power, 2,000.

STATEMENT OF OPERATIONS STEAMER *MANOLA*, SEASON OF 1890.

Receipts.	Amount.	Per ton freight carried.
Freights	$93,738.25	$1.317
Disbursements.	**Amount.**	**Per ton freight carried.**
Fuel	$14.427.14	$0.203
Oil	863.82	0.012
Machinery and boiler repairs and supplies	1,182.54	0.017
Cabin supplies and provisions	3,304 98	0.046
Handling cargoes	16,111.75	0.226
Hull repairs	149.34	0.002
Towing	744.00	0 011
Chandlery	1.328.37	0.018
Sundry sources	886.24	0.012
Labor	10,021.37	0.141
Outfit	623.44	0.009
Insurance	5,471.21	0.077
Total expenses	$55,114.20	$0.774
Net earnings	$38,624.05	$0.543

Disbursements.	Amount.
Gross earnings per day	$425.24
Operating expenses per day	248.26
Net earnings per day	173.98

Percentage of operations to earnings	58.79
Earnings per ton per mile	0.00078
Operating expenses per ton per mile	0.00046
Net earnings per ton per mile	0.00032
Earnings per mile traveled	1.853
Operating expenses per mile traveled	1.090
Net earnings per mile traveled	0.763
Total miles traveled	50,584.
Average miles per day	227½.
Tons freight carried	71,170,690 tons.
Tons freight carried one mile	3,600,078,861 tons.
Average speed per hour, light	12.72 miles.
Average speed per hour, loaded	11.85 miles.
General average speed per hour	12.25 miles.
Total tons fuel used	5,528 tons.
Average tons fuel used per trip	184.553 tons.
Average amount per mile, light	209 pounds.
Average amount per mile, loaded	226 pounds.
General average amount fuel per mile	218 pounds.
Average fuel per ton per mile	1½ ounces.
Number of trips	30.
Average size cargo	2,295 82 tons.
Average draft water Sault canal	14.7–14.9 feet.
Average time loading	7½ hours.
Average time unloading	12 hours.
Average time handling cargo	19½ hours.
Average tons loaded per hour	306.244 tons.
Average tons unloaded per hour	191.712 tons.
Average tons handled per hour	235.47 tons.
Actual time sailing	175 days.
Actual time in port	47 days.
Actual time in commission	222 days.
Per cent. of time sailing	78.83
Per cent. of time in port	21 17
Average number of crew each trip	23
Average wages crew each trip	$334 05.
Average length of trip	7 days 9½ hours.
Average mileage per trip	1,686

Coal, short tons; cargo, long tons.

COMMERCE ON THE GREAT LAKES.

[From Report of S. G. BROCK, Chief of the Bureau of Statistics, Treasury Department, Washington, D. C.]

The Great Lakes, with their connecting waterways, have a water surface of 95,275 square miles, a watershed of 174,800 square miles, the two combined making a basin area of 270,075 square miles.

The coast lines of this great lake system contain more than half of the fresh water area of the globe, and have a combined length in the United States of 3,075 miles.

The distance from Ogdensburg, the eastern point of navigation on the United States coast line, to Duluth, the extreme western point of Lake Superior, is 1,279 miles, and the distance is about 500 miles in a direct line from the extreme north shore of Lake Superior to the southern coast line of Lake Michigan. Within these large areas and coast lines are inland waterways that are not equaled upon any other continent.

The immense advantages of these great waterways and great facilities offered for traffic will be appreciated when it is understood that eight large States, with abundant and varied resources, border on these coast lines, having, according to the census of 1890, an active and thrifty population of 26,029,533, and there are thirty-six towns and cities upon the banks of these lakes and channels, each having a population exceeding 1,000. Six of these cities have a population each in excess of 100,000 and aggregating a population of 2,161,624.

The first American vessel upon Lake Erie was built at Erie in 1797. The first steam vessel that navigated the lakes was built at Sacket's Harbor, New York, in 1817, and measured 240 tons. She was named *Ontario.*

The *Walk in the Water* was the first steamer constructed above Lake Ontario, and was launched at Black Rock in May, 1818.

This steamboat navigated regularly between Black Rock, on the Niagara river, and Detroit. Soon after followed the steamers *Superior*, *Pioneer* and others, up to 1840, during which time propellers had been built, a new method devised by the great inventor, Ericsson, and which proved very successful. There was now a rapid growth in the commerce of the lakes. The schooner *Illinois*, of 100 tons burden, was the first vessel to arrive at Chicago from the lower lakes. "This important event occurred July 12, 1834, when she was hauled over the bar. On that occasion all the male inhabitants of the village, including the boys, amounting to nearly one hundred, assisted in dragging the craft across the bar. Flags were then raised on the schooner, and she sailed up the forks of the river in fine style."

At this time the commerce was mostly to the West, the passenger traffic was the most lucrative, and, singular as it will now appear, the cargoes were chiefly of flour, provisions, and supplies for the settlements that were constantly extending westward. In 1840 the grain movement from the West was well established, and has continued to grow until it has reached its present splendid proportions.

The navigation of Lake Superior commenced much later and the development of its commerce was much slower than on the lower lakes. The first American vessel launched upon Lake Superior was the *John Jacob Astor*, in 1835. This was soon followed by the construction of other vessels by the American Fur Company in 1837. In 1845 new impulse was given to the building of vessels and the navigation of this greatest of all the lakes by the discovery of copper on its shores. In the latter part of 1845 the propeller *Independence*, the first steam vessel, of 260 tons burden, was launched.

GROWTH OF THE LAKE FLEET.

The growth of the lake fleet from 1849 has been rapid and surprising. The total tonnage of the shipping on the Great Lakes in 1849 was 161,832 tons, valued at $7,868,000, and was made up of 60,752 tons in steam vessels and 101,080 tons in sailing vessels. In 1862, or thirteen years subsequently, there were navigating the waterways of these great lakes 350 steam

vessels with a measurement of 125,620 tons, of the value of $5,482,900, and 1,152 sailing vessels with a tonnage of 257,689 tons, of the value of $6,379,550, or a total tonnage of 383,309 tons and a total value of $11,862,450.

It will be observed that in the thirteen years from 1849 to January, 1862, there was an increase of 136.81 per cent in the amount of tonnage and of 50.77 per cent. in the value of the lake fleet.

The growth of the Lake fleet during recent years has been still more remarkable and gratifying. According to the Inland Lloyd's Register for 1886 the business fleet of the lakes on September 1, 1886, consisted of 1,997 vessels, with a tonnage of 634,652 tons and of the value of $30,597,450. From 1862 to 1886, it will be observed, there was an increase of 65.57 per cent. in the tonnage, and of 157.93 per cent. in the value of the business fleet of the lakes. From the same authority it is learned that on December 1, 1891, the number of vessels had increased to 2,125, the tonnage to 870,981 and to a value of $57,054,750. This is exclusive of small tugs, steam canal boats and sailing vessels, which are strictly not a part of the carrying fleet. Including these, however, the total registered, enrolled and licensed tonnage of all craft afloat on the Northern Lakes in 1891 was 1,154,870 tons.

Thus it will be seen that in the five years from 1886 to December 1, 1891, there was an increase in the tonnage of 236,329 tons, or 37.24 per cent., and of $26,457,300, or 86.47 per cent., in the value of these immense inland shipping interests, which are the wonder and admiration of the commercial world.

The value of the steam vessels in 1886 was $22,047,200, and in 1891 the total valuation of steam vessels was $49,543,750, an increase in the value of $27,496,550.

VALUE AND SIZE OF VESSELS.

Still more remarkable and surprising is the advance and improvement made in the size and material of which the steam vessels are constructed.

In 1859 the largest propellers on the lakes averaged about

700 tons net register, varying from 580 to 980 tons. The register of 1886 shows there were 21 steamers on the lakes which had a net registered tonnage to exceed 1,500 tons, and in 1891 the register shows there were 126 of such steamers. December 1, 1886, there were on the Great Lakes only 6 steel vessels, with a tonnage of 6,459 tons, of the value of $694,000; and December 1, 1891, there were 89 steel vessels, with a tonnage of 127,624 tons, and with a valuation of $14,502,500. Of the 89 vessels all but 19 are steamers, and their average value exceeds $190,000. The rapid advance being made in the lake commerce will be still further appreciated when it is stated that 45 new vessels are now contracted for or being constructed, all to be ready for the season 1892. Of these, 40 are steamers and 32 are of steel. The value of these 45 new vessels, which make a considerable fleet of themselves, is estimated at $4,896,000.

At the same time the average carrying capacity of the largest coarse-freight steamers has in one year increased from 2,652 to 3,200 tons.

COMPARISON WITH OCEAN VESSELS.

In this connection it will be instructive to make a comparison of the lake vessels and their tonnage with the vessels and their tonnage engaged in the foreign and coastwise trade on the Atlantic, Gulf and Pacific coasts. Such a comparison will furnish forcible evidence of the amount and value of shipping interests of the inland waterways. The number of American vessels engaged in the foreign trade on the Atlantic, Gulf and Pacific coasts combined was 1,579. On the northern lakes the total fleet was 3,600 or 2,021 in favor of the latter. The total tonnage of the American vessels engaged in the foreign trade on the Atlantic and Gulf coasts in the fiscal year 1891 was 781,655 tons.

The total tonnage of American ships on the Pacific coast engaged in the foreign trade was 217,592 tons. The total tonnage of vessels on the northern lakes was at the same time 1,154,870 tons, or 373,215 tons in excess of the Atlantic and Gulf coast tonnage, and 937,278 tons in excess of the tonnage

of the Pacific coast vessels engaged in the foreign trade. The total tonnage of vessels engaged in the coastwise trade on the Atlantic and Gulf coasts in 1891 was 1,999,028 tons, and upon the Pacific coast it was 223,265 tons, as compared with 1,154,870 tons, the total tonnage of the fleet of the northern lakes.

The total tonnage of the vessels, foreign and coastwise, of the Atlantic, Gulf and Pacific coasts in 1891 was 3,221,541 tons.

It will be observed the tonnage of the vessels upon the northern lakes was over one-third of this combined tonnage of the Atlantic, Pacific and Gulf coasts, or 35.85 per cent. of the same. The total tonnage of the entire merchant marine of the United States in 1891, including all the seaboard, the rivers and lakes, was 4,684,759. It appears, then, the tonnage of the Great Lakes is about one-fourth of the whole.

TONNAGE OF CANADIAN VESSELS ON THE GREAT LAKES.

The number of vessels and number of tons on the registry books of the Dominion of Canada for the Province of Ontario on December 31 in each year from 1873 to 1891, both inclusive, were as follows: 1873, 681 vessels, of 89,111 tons; 1874, 815 vessels, 113,008 tons; 1875, 825 vessels, 114,990 tons; 1876, 889 vessels, 123,947 tons; 1877, 926 vessels, 131,761 tons; 1878, 958 vessels, 135,440 tons; 1879, 1,006 vessels, 136,987 tons; 1880, 1,042 vessels, 137,481 tons; 1881, 1,081 vessels, 139,998 tons; 1882, 1,112 vessels, 137,061 tons; 1883, 1,133 vessels, 140,972 tons; 1884, 1,184 vessels, 142,387 tons; 1885, 1,223 vessels, 144,487 tons; 1886, 1,248 vessels, 140,929 tons; 1887, 1,275 vessels, 139,548 tons; 1888, 1,330 vessels, 139,502 tons; 1889, 1,352 vessels, 141,839 tons; 1890, 1,312 vessels, 138,738 tons, and 1891, 1,345 vessels, 138,914 tons. Number of vessels, 1891, 1,345; number of steamers, 1891, 741; gross tonnage of steamers, 1891, 92,785—total registered tonnage, 1891, 138,914.

From the above statement it appears there has been a steady growth in the number and tonnage of the Canadian vessels upon the Great Lakes system, but not as large as in the United States lake fleet, as will appear by comparison. The number of registered vessels in the business fleet of the United States on the

Great Lakes, December 1, 1891, was 2,125, and with a tonnage of 870,981 tons. If with these, however, were included small tugs, steam canal boats and sailing vessels, strictly not included in the carrying fleet, the total registered and licensed tonnage on the Northern Lakes in 1891 amounted to 1,154,870 tons.

COMPARATIVE SIZE OF VESSELS.

A still more surprising contrast is the size of the vessels in the fleet of the lakes as compared with the vessels of the seaboard, the average size of the vessels composing the fleet of the northern lakes being considerably greater than of the vessels of the seaboard.

It appears from the records of the Bureau of Navigation* that the sailing vessels of the seaboard average 128 tons, and the sailing vessels of the lakes 258 tons. The steam vessels of the seaboard average in measurement 299 tons, and of the lakes 428 tons. Singular as it may seem, nevertheless it is a fact, that the steam vessels of the lakes are not only larger, but there is a greater number of large steamers than in all the balance of the country. There are 272 steam vessels on the lakes, with a tonnage of from 1,000 to 2,500 tons, and an aggregate tonnage of 439,787 tons. On the seaboards and rivers there are 207 steam vessels of the measurement mentioned, with an aggregate tonnage of 308,694 tons, or an excess in favor of the lakes of 65 steamers in number, and of 131,093 tons in tonnage. It furthermore appears that the average tonnage of the steam and sailing vessels on the lakes is 349 tons, and on the seaboard 165 tons, or more than double the latter.

From these facts it appears that not only has the growth of the lake fleet been rapid, but the character of the structure and material of this fleet has greatly changed and improved so as to reflect great credit upon the enterprise of the builders and owners of the floating equipment of the Great Lakes.

During five years the sailing vessels have decreased, but the steam vessels have increased, and the tonnage of each class has increased.

* Treasury Department.

In 1886 there were 1,060 sailing vessels, of an aggregate tonnage of 309,767 tons. In 1891 there were 888 sailing vessels, of the aggregate tonnage of 310,393 tons. In 1886 there were 937 steam vessels, with an aggregate tonnage of 324,885, and valued at $22,047,200. In 1891 there were 1,237 steam vessels, with 560,388 aggregate tonnage, and a valuation of $49,543,750. Thus, there was an increase of 32.02 per cent. in the number, of 72.49 per cent. in the tonnage, and of 124.72 per cent. in the valuation within the five years from 1886 to 1891.

Of the 89 steel vessels, 70 were steam vessels. Something of the character and carrying capacity of these modern steel steamers may be understood when it is stated that one of these, the *E. C. Pope*, in the season of 1891 transported from Chicago to Buffalo 125,990 bushels of corn, the largest cargo of grain that has been carried on the lakes. The weight of the cargo was 3,527 tons. It is further stated that the largest cargo ever carried on the northern lakes was on the steel steamer *Maryland*, which is similar in construction and dimensions to the *E. C. Pope*. This cargo was iron ore of 3,720 tons and was carried from Escanaba to South Chicago. This same steamer in the season of 1890 carried twenty-nine cargoes of iron ore, averaging 3,581 tons each.

SHIPBUILDING.

A very good understanding of the rapid progress being made in the equipment of the fleet of the Great Lakes may be had by observing the great increase in shipbuilding on the lakes during recent years. In the following table is a statement of the vessels built and their gross tonnage during the five years from 1887 to 1891, inclusive:

	1887. Tons.	1888. Tons.	1889. Tons.	1890. Tons.	1891. Tons.	Total five years.
Entire seaboard.....	83,061	105,125	111,852	169,091	237,462	706,501
Northern lakes.....	56,588	101,103	107,080	108,526	111,856	485,053
Western rivers.....	10,901	11,859	12,202	16,506	19,984	71,452

It will be observed that in the five years the new vessels built upon the lakes were over one-third of the whole and over one-half, or 62.34 per cent., of what was built in the balance of the whole country.

If we consider steam tonnage built during the same five years, the comparison will be found still more favorable to the northern lakes. A statement is given in the following table of the steam vessels and tonnage built on the seaboard, lakes and rivers for the period mentioned.

	1887. Tons.	1888. Tons.	1889. Tons.	1890. Tons.	1891. Tons.	Total five years.
Atlantic and Gulf coasts.....	38,972	30,466	41,308	54,241	67,074	232,061
Pacific coast.....	3,750	12,710	12,747	6,896	9,167	45,270
Northern lakes.....	47,183	87,459	93,707	86,023	93,323	407,695
Western rivers.....	10,168	11,372	11,557	11,886	15,473	60 456

It will be observed that the steam tonnage built on the northern lakes during the five years mentioned exceeds that built on the Atlantic and Gulf coasts by 175,634 tons, and exceeds the steam tonnage built in all the balance of the United States by 69,910 tons. In 1890 there were 116 steam vessels built on the northern lakes, with an average tonnage of 741.5 tons. In 1891 there were 123 steam vessels built, with an average tonnage of 768.7. Thus the movement in shipbuilding on the Great Lakes seems to be in the direction of large steam vessels with greater carrying capacity.

In the following table is a statement of the tonnage of the vessels on the seaboard, northern lakes and western rivers for eleven years from 1881 to 1891, inclusive:

Year ending June 30.	Atlantic and Gulf. Tons.	Pacific. Tons.	Northern lakes. Tons.	Western rivers. Tons.	Total. Tons.
1881.....	2,713,918	286,385	663,382	394,049	4,057,734
1882.....	2,759,220	302,435	711,369	493,009	4,166,033
1883.....	2,822,962	327,565	723,912	361,047	4,235,486
1884.....	2,847,135	334,669	733,069	356,355	4,271,228
1885.....	2,809,323	360,609	749,498	346,054	4,265,934
1886.....	2,685,988	347,685	762,560	334,902	4,131,135
1887.....	4,638,273	356,445	783,722	327,405	4,105,845
1888.....	2,612,841	399,890	874,103	305,082	4,191,916
1889.....	2,599,504	436,273	972,271	299,427	4,307,475
1890.....	2,638,596	427,392	1,063,064	295,446	4,424,498
1891.....	2,780,683	440,857	1,154,870	308,347	4,684,757

It will be discovered there has been a constant growth in the tonnage of the northern lakes, and it has nearly doubled in the time given, while there has been little increase in the tonnage

on the seaboard, and a decrease in the registered tonnage of the western rivers. There has, however, been a large increase in the number and tonnage of unrigged crafts.

ELEVATOR AND DOCK FACILITIES.

The remarkable increase in the transportation facilities in recent years, and which appears in the splendid fleet equipment of the Great Lakes, and the surprising growth of the lake commerce, which has attained to such wonderful proportions, have required much greater improvements in the means and methods of handling the great cargoes that come into and go out of the numerous ports.

As the commerce of the lakes has increased there has been a corresponding improvement in elevator and dock facilities to meet the demands for rapid and efficient service in handling the cargoes. These facilities are unsurpassed at any other ports of the world.

Following is given the number and storage capacity of the grain elevators at the leading ports on the Great Lakes: Chicago, 26 elevators, 28,675,000 bushels; Duluth and Superior, 17 elevators, 21,200,000 bushels; Milwaukee, 10 elevators, 5,430,000 bushels; Toledo, 13 elevators, 7,357,000 bushels; Buffalo, 32 elevators, 15,000,000 bushels; Detroit, 4 elevators, 3,100,000 bushels; Port Huron, 6 elevators, 1,670,000 bushels; Cleveland, 1 elevator, 650,000 bushels.

The only port in the United States which approximates to Chicago in its facilities for handling grain is New York, which has twenty-seven elevators, with a capacity of 27,275,000 bushels. Some of these elevators at the lake ports can remove the cargo of grain from a vessel at the rate of 25,000 bushels per hour. The facilities for handling grain have been constantly improved, and have resulted in great saving of time and labor.

Great improvement has also been made in providing dockage and in methods of handling coal, ore and other merchandise, so that where formerly vessels were compelled to lie in harbor days during the discharge and taking on of cargoes, now only a few hours are required, and they are quickly released to renew

their voyage. At this time there is a coal dockage at the nine leading shipping ports upon Lake Erie of 38,272 feet in length, with an average width of 158 feet, and a daily capacity of 75,000 tons. At Two Harbors, Duluth, West Superior and Ashland there is a dockage of 4.370,875 square feet, with a storage capacity of 2,270,000 tons and a daily handling capacity of 37,600 tons. Likewise admirable facilities have been provided for handling iron and copper ores, stone, lumber, salt, etc Cargoes of coal and ore of from 2,500 to 3,500 tons are loaded in two or three hours and unloaded in from six to twelve hours.

In addition to the extensive harbor and wharfage facilities that have been provided for the immense traffic that now exists in grain, coal, iron ore, etc., large warehouses have been built to meet the great demands for storage of flour and miscellaneous merchandise where one or more of the great lake vessels can immediately discharge their cargoes.

THE THORNYCROFT BOILERS OF THE DANISH CRUISER *GEISER*.

[Reprinted from "Engineering."]

The cruiser *Geiser* was selected some time ago by the naval authorities of Denmark as a suitable vessel for the purpose of trying the value of the Thornycroft water-tube boiler on a larger scale—*i. e.*, with a larger number of boilers in combination—than had hitherto been used. The Danes have already had considerable experience with this boiler, but it has been in smaller craft, and the main problem required to be solved was whether the working would be equally successful in larger groups, the greatest number of boilers hitherto tried, working in one vessel, being two. Judging from the report of the trial which is before us, the answer is distinctly in the affirmative, the results having been perfectly successful. It would be difficult to overestimate the importance of this matter at the present time, when we are having so much trouble with the ordinary boilers in our own navy. It seems almost as if finality had been reached in boiler construction on the old lines. Forced draft and high pressures are very hard on the shell boiler, and these high pressures are necessary if the full value is to be got out of the multiple-compound engine. In fact the engine is, as it always has been, ahead of the boiler. Those who support the water-tube principle say this is because of the vicious system, originated by the first boiler-makers, of generating steam in a pot instead of a pipe. However this may be, the pipe boilers for long brought forward—and there has been no lack of numbers or variety, dating back from the earliest days onward—have generally possessed such serious disadvantages that they have been unsuitable for marine purposes. It was reserved for Mr. Thornycroft to overcome the difficulties which beset the path of the water-tube boiler designer, for we think that we may fairly say he has achieved success,

considering the good results obtained already, and the time the boiler has been in satisfactory use in many cases. It may be stated here that the first Thornycroft boiler made, after working in a boat for six years, was taken out and is now working a saw-mill engine. It is not our present intention, however, to review the history of the Thornycroft boiler and water-tube boilers in general, as the matter has been dealt with in former pages of "Engineering," and we will therefore proceed to deal with the particular examples under notice.

The *Geiser* is a third-class cruiser of about 1,300 tons displacement, and was built at Copenhagen. She was launched in 1891, and has a protective deck. She is about 257 feet 6 inches long and 27 feet 6 inches beam. The estimated indicated horse-power was 3,000, which, it will be seen, the engines exceeded by 157 horse-power on the forced-draft trial. The armament consists of two 4.7-inch quick-firing guns and four 57-centimeter guns. There are also four launching carriages for Whitehead torpedoes. There are two sets of vertical triple-expansion engines, which were made at Messrs. Burmeister & Wain's works at Copenhagen. The cylinders are 19-inch, 30-inch and 50-inch by 18-inch-stroke. The vessel is twin-screw, there being two Thornycroft propellers having forged blades. The diameter of each screw is 8.5 feet and the pitch 7.59 feet. It is, however, with the boilers that we are mostly concerned. There are eight Thornycroft boilers, arranged in two boiler-rooms, with four stokeholds and two funnels. These boilers were made at the Royal Dockyard, Copenhagen, under license granted by Messrs. Thornycroft & Co. The total tube surface is 12,000 square feet, and the total grate area 171 square feet. These boilers are of the same general type as illustrated in "Engineering," Vol. xliv, page 104, where the particulars of the boilers of the Spanish torpedo boat *Ariete* are given.* A novelty has been introduced in this boiler in regard to the material of the tubes, steel having been used in the outer and inner rows, and brass in the intermediate rows. Brass tubes had formerly been tried by Messrs. Thornycroft, but not with such a measure of success as led to the material being used permanently for all the tubes.

* See also JOURNAL, Vol. II, p. 176.

The steam trials took place in the Sound at Copenhagen, and consisted of a six-hours' coal-consumption trial, an eight-hours' sea-speed trial, and a four-hours' full-power trial. The mean results obtained are given in the following table:

	Coal consumption trial.	Sea speed trial.	Full power trial.
Date of trial	Oct. 27.	Nov. 2.	Nov. 5
Duration of trial ..... hours...	6	8	4
Displacement of ship on trial..... tons...	1,265	1,259	1,276
Steam pressure in boilers..... lbs...	167.0	176.0	177.6
Steam pressure in high pressure valve chest.. lbs...	146.6	164 9	164.0
Steam pressure in first receiver. ..... lbs...	56 6	70.0	75.0
Steam pressure in second receiver..... lbs...	5.4	9.5	13.0
Vacuum..... inches...	26.5	25.3	25.1
Air pressure..... inches of water	Natural draft.	0.57	0.81
Revolutions of fans per minute.....	...	475	571
Ratio of cut-off.....	0.48	0.54	0.68
Mean pressure in high-pressure cylinders... lbs...	33.6	46.4	52.2
Mean pressure in intermediate pressure cylinders..... lbs...	24.7	30.8	36.4
Mean pressure in low-pressure cylinders.... lbs...	10 0	12.0	14.6
Mean revolutions per minute of main engines...	204.7	227 9	250.6
Collective indicated horse power.....	1.744	2,422	3,157
Mean temperature in forward funnel.... deg. F...	423	466	585
Mean temperature in aft funnel..... deg. F...	439	482	628
Mean temperature between boiler tubes:			
Stokehold I (foremost), Starboard.. deg. F...	664	882	970
Stokehold I (foremost), Port..... deg. F...	727	896	1,078
Stokehold II..... Starboard.. deg. F...	752	934	963
Stokehold II..... Port..... deg. F ..	682	851	970
Stokehold III..... Starboard.. deg. F...	639	815	876
Stokehold III..... Port..... deg. F...	670	828	901
Stokehold IV (after).... Starboard.. deg. F...	631	804	932
Stokehold IV (after).... Port..... deg. F...	729	919	1,004
Total coal consumption on trial:			
Stokehold I..... Starboard ..... lbs...	2,260	4.850	3,160
Stokehold I..... Port..... lbs...	2,160	4.900	3,160
Stokehold II..... Starboard... ... lbs...	2,360	4,390	3,020
Stokehold II..... Port..... lbs...	2.340	4.010	2,810
Stokehold III..... Starboard ..... lbs...	2,320	4,170	2,780
Stokehold III..... Port..... lbs...	2.210	4.040	2,760
Stokehold IV..... Starboard ..... lbs...	2,310	3.940	2,900
Stokehold IV..... Port..... lbs...	2,310	4,270	2,990
Coal consumption per hour in all boilers... lbs...	3,045	4.309	5,895
Coal consumption per hour per indicated horse power.. ..... lbs...	1.75	1.77	1.87
Coal consumption per square foot grate area, lbs...	17.8	25.2	34.5
Speed of ship..... knots...	14.34	16.00	17.1

The temperatures referred to were measured by means of graphite pyrometers, one at the base of each funnel, and one between the boiler tubes on one side of each boiler.

The report of the Danish director of naval construction, Captain Nielsen, states:

"During the trials the boilers worked most excellently; steam was kept up with the greatest ease. The steam-generating power could be regulated nearly immediately to suit the steam consumption of the engines, by means of the stop-valves on the steam pipes of the fans. At the end of the sea-speed trials we forced the engines up to 3,314 indicated horse power in a few moments, after having steamed over 7½ hours, with an air pressure of about 0.6 inch, and could keep the steamer at this high horse-power with an air pressure about 1 inch. When going full speed we often stopped the ship immediately, and we could bring the ship up to full speed in a few minutes, it being not necessary to pay any regard whatever to the boilers, these being able to stand all sudden changes of temperature. The boilers did not prime either during the highest forcing nor during sudden changes in the working of the engines."

English naval engineers will read these statements with envy. When we remember how careful they have to be in regulating the steam pressure in ordinary boilers, so that the control of the draft becomes one of the most critical things which the engineer has to attend to, such perfect immunity from trouble revealed by Captain Nielsen's statements seems something too good to hope for. It may be contrasted with the attempt to run the forced-draft trials of H. M. S. *Jason*. This vessel is a first-class torpedo gunboat, a craft somewhat smaller than the *Geiser*, being 230 feet long by 27 feet wide. She was taken out of Sheerness on the 13th inst., in order to make 19 knots with 3,500 indicated horse-power. The trial is reported to have been unsatisfactory, it is said owing to priming.

It will be noticed that the funnel temperatures are not high, averaging about 474 degrees Fahr. for the forced draft and 606 degrees Fahr. for the full-power trial. The Danish engineers are to be congratulated in getting such a good result with these

moderate conditions, which are well reflected in the coal consumption, the rate of 1.87 pounds of coal per indicated horse power per hour being a wonderful performance for a forced-draft trial, even though the air pressure did not reach one inch of water.

Some instructive figures are also given on the important point of weight. The official report says: "Another great advantage which the boiler possesses is its small weight, as compared with ordinary boilers. This can best be seen by comparing the weight of the boilers in the *Geiser* with the weight of the boilers in the cruiser *Hekla*. This cruiser has exactly the same engines as the *Geiser*, and its six low cylindrical boilers are proportioned to the same horse power (3,000 indicated horse power) as the *Geiser*, both ships being of the same type and displacement.

	Hekla. Tons.	*Geiser.* Tons.
Boilers with pipes, feed pumps, uptakes, funnels and all other fittings in the boiler rooms	120.2	90.8
Water in boilers	48.0	17.4
Totals	168 2	108.2

Thus 60 tons, or over one-third of the total weight of the boilers in the *Hekla*, are saved in the *Geiser*.

Looking at the matter in another way, it will be seen that 29.177 indicated horse power is got from each ton of boiler and water in the case of the *Geiser*. In the *Australia* and *Galatea* only 22 indicated horse power were obtained per ton weight of boiler and water. It may be said that it is not fair to compare a return-tube boiler with the boilers of a small, fast vessel, and that it would make a fairer comparison to select the light locomarine boiler of the torpedo-boat type. We do not think this line of argument can be upheld, in view of the low coal consumption; supposing, of course, the Thornycroft boiler to possess substantial seagoing qualities fitting it for the continuous work of ocean voyages for periods of time approximately corresponding to those of the return-tube boiler. Our readers will, of course, recognize that a comparison between the machinery of a small vessel like the *Geiser* and a large cruiser such as the *Australia* or *Galatea* is hardly fair to the former, as the ratio

of weight of fittings, &c., and other boiler-room weights must be higher with the smaller power. Perhaps we may better illustrate this by taking the case of the *Speedy*, the torpedo-boat now being built by Messrs. Thornycroft. She is to have the water-tube boiler of the same class; the weight of machinery, which is estimated to develop 4,500 indicated horse-power, is practically the same as required in other vessels of her class for an estimated power of 3,500 indicated, the space occupied being the same. We shall look forward with much interest to the development of the figures by trial. The *Jason*, already referred to, is one of the vessels of this class. Generally it may be said that in the matter of space occupied the Thornycroft boiler has been conclusively proved to stand on as good a footing (at least) as the older types. The chief advantages in this boiler are obtained by the fact that a large area of heating surface can be combined within small dimensions of space and weight. There can be no doubt that the endeavor to cut down the weight and space is at the root of all the recent boiler troubles in the Navy. In the case of the *Geiser* it will be seen that about 4 square feet of heating surface are allotted to each indicated horse-power. In the case of many of our war vessels the effort has been made to reduce the heating surface to less than one and a half square feet to each indicated horse-power. This must lead to waste of fuel, and has led to disaster in other respects.

We will conclude this notice with the following further extract from the official report:

"The great ease with which steam is formed in these boilers was most apparent on a trial, at which we tried how quick steam could be raised. The temperature of the boiler water was 13 deg. C. when fires were lighted and the gauge glasses three-quarters full.

	h.	m.
Fires were lighted at	8	30
Centrifugal pumps and fans were started with 8 lb. pressure in boilers at	8	54
Steam pressure, 40 lb. per square inch, at	9	4

The following letters to *Engineering* contain valuable information with regard to the relative weights of locomotive and tubulous boilers as used on small vessels:

THE DANISH CRUISER *GAISER*.

To the Editor of "Engineering."

Sir: With great interest I have read your article in "Engineering," December 23, about the trial of the Thornycroft boiler on board the Danish cruiser *Gaiser*.

As for all connected with the progress of marine engineering, the question of steam boiler construction is at present a very interesting one, I hope the following data and comparison may be of some value.

Some long time already I have been trying to get a comparison made between a good system of water-tube boiler and the locomotive boilers built by Mr. Schichau, at Elbing, for similar purposes. Up till now, I never could get any trustworthy information about the water-tube system, and I have only to thank the courtesy of Messrs. Thornycroft, in publishing these results, that I am at last able to do so.

I hardly need say that there is no other feeling than real technical interest on my part in making this comparison, and as I esteem Mr. Thornycroft as the foremost and most able constructor of the system of water-tube boilers for marine purposes, a comparison with him and that which other people have achieved on other lines can only prove of value.

I firstly take the data as published by you: The *Gaiser* is a ship of 1,276 tons displacement, has two engines (triple-expansion) 18 inches (460 mm.) stroke by 19 inches, 30 inches and 50 inches (480 mm., 760 mm. and 1,270 mm.) cylinder diameter; eight water-tube boilers, arranged in two boiler rooms, with four stokeholds and two funnels.

Steam pressure at full power, 177.6 pounds (about 12 atmospheres); revolutions of engines, 250.6 per minute, and total indicated horse power, 3,157.

	Tons.
Weight of boilers with pipes, pumps, uptake, funnel and all other fittings in boiler room	90.8
Water in boilers	17.4
Total	108.2

That is 34.3 kilogrammes per indicated horse power.

The cruiser type *Kassarsky* or *D* class, of which Mr. Schichau, at Elbing, has built about ten, are ships of 400 tons displacement, and have one triple-expansion engine of 23⅝-inch (600 mm.) stroke by 27½-inch, 44⅛-inch and 66¼-inch (690 mm., 1,120 mm., and 1,680 mm.) cylinder diameter, and two locomotive boilers in one boiler room, one stokehold and one funnel; steam pressure, 12 atmospheres; revolutions per minute, 250, and indicated horse power 3,500.

As you will see from the detailed list of weights given below, the weight of engine complete, with shafting, screw and with boilers, water, all auxiliary engines, floor plates, supplementary and spare gear complete, is 112.7 tons—that is, 32.2 kilogrammes per indicated horse power.

That means the water-tube boilers on the *Gaiser* weigh alone more than on the *Kassarsky*-type engines and boilers together.

The boilers alone, with water and all fittings in boiler room, weigh, on the *Kassarsky* type, only 19.2 kilogrammes per indicated horse power.

These types of engines and boilers have been in constant work now about six years, and have given entire satisfaction. The best proof of this probably is that after the delivery of the first of these ships, the German, Russian and Austrian governments have ordered and received several more exactly similar vessels. The boilers have never given trouble, never prime, and the engines have proved excellent and up to any requirement.

The coal consumption of the *Gaiser*, with about 1-inch air pressure, is stated to be 1.87 pounds (about 0.9 kilogrammes) per indicated horse power per hour, whereas on the *Kassarsky* type it does not exceed 0.75 kilogrammes per indicated horse power per hour on full power.

About the space taken up by the eight water-tube boilers I am not able to judge without drawings and dimensions, but I believe I shall not be very far out if I take it to be at least double that required for the two locomotive boilers of the *Kassarsky* type.

If we are able to achieve such results with our old friend, the locomotive boiler, in every-day practice, I may well ask if there

is any necessity to go off from these lines, before at least somewhat similar results have been reached with other systems.

List of Weights of Engines and Boilers for Cruisers, Type *Kassarsky*, built by Mr. F. Schichau, Elbing; 3,500 indicated horse power.

		Kilogrammes.	
	A.		
1.	Boilers with clothing	40,367.50	
2.	Fittings	6,458.70	
3.	Uptake and funnel	4,692.10	
4.	Water in boilers	15,800.	
	Total A		67,318.30
	B.		
1.	Main engine, complete with condenser, air, feed, and bilge pumps, tanks, evaporator, Kingston's tubes, &c	40,458.01	
2.	Water in condenser, pumps and tubes	2,284	
3	Shafting, stern tube, screw and bearings	8,874	
	Total, B		41,616.01*
	C.		
1.	Ventilating engines and ventilator complete	663.20	
2.	Steam pumps complete	405	
3.	Circulating pump and engine complete	540.50	
	Total, C		1,608.70
	D.		
1.	Floorplates in boiler and engine room and telegraph complete with gear	1,064.58	
	Total, D		1,064.58
	E.		
1.	Spare gear	343.25	
2.	Supplementary	752.10	
	Total, E		1,095.35
	Total		112,702.94*

I am, sir, yours very truly,

R. A. ZIESE.

ST. PETERSBURG, *December* $\frac{17}{29}$, 1892.

THE BOILERS OF THE DANISH CRUSIER *GAISER*.

To the Editor of "Engineering."

Sir: We have read with great interest Mr. Ziese's letter on the trials of the Danish crusier *Gaiser*, but we fear that in his

* Error of 10,000 kilogrammes in addition; see p. 172.

eagerness to make a comparison, like the "guid man" in the song, he has seen a comparison where no comparison should be.

The *Gaiser* was built, as a crusier, of more than three times the displacement of the type *Kassarsky*, and it was not by any means intended that she should develop the utmost power that could be got out of the Thornycroft water-tube boiler.

The machinery was originally designed to develop 3,000 indicated horse-power, and the fact that this was exceeded on the four hours' trial with a coal consumption of only 34½ lbs. per square foot of grate, shows that the Danes, who designed the boilers, had determined that they should obtain the power required by very easy steaming, which, of course, means larger and heavier boilers than would otherwise have been required.

These conditions are very different from those which obtain in torpedo boats, where the utmost power is exacted from the smallest amount of weight.

We may mention with regard to weights that the boiler-room contents of several of our boats, fitted with water-tube boilers, averaged only some 15 kilogrammes per indicated horse-power, and that these boiler-room contents included piping, cocks and valves, donkeys and spare gear, which Mr. Ziese does not include in his group A, although he uses the total of that group in calculating the 19.2 kilogrammes which he gives as the boiler-room weights per indicated horse-power in the *Kassarsky*.

We may say, further, that these results were obtained with only from 2 inch to 2½ inch of wind pressure, and that so completely was the command of steam in the control of the engineer that, as in the case of the *Gaiser*, the steam pressure could be regulated by simply turning the valve on the steam pipe of fan engine, and all this without leakage or priming.

We fancy these facts will tend to shake Mr. Ziese's confidence in his old friend, the locomotive boiler, even of his latest and best type.

To pass to another matter, we note that Mr. Ziese has made an error of 10,000 kilogrammes in the addition of group B, and that he has carried this error through to his grand total, so that the weight he gives ought to be 122,702.94 instead of 112,702.94

kilogrammes, as given, and that in the whole list of weights there is no mention of the piping or cocks and valves.

Possibly these and some other omissions, doubtless owing to Mr. Ziese's being away from Elbing, may account for the weight given for this machinery of 3,500 indicated horse power being less than that of the vessels delivered to one, at least, of the Governments named by Mr. Ziese subsequently, as he says, to the delivery of the first of the ten ships of the class built by Mr. Schichau, and which, we understand, weighed over 130 tons, and only developed 3,000 indicated horse power.

Certainly, unless there is some further omission in the list of weights, which we are not in a position to indicate, Mr. Schichau is to be congratulated on getting 3,500 indicated horse power in every-day practice out of 67,318.30 kilogrammes of boiler-room weights, seeing that, in the case of the *Sharpshooter*, 3,500 indicated horse power was barely obtained with a specified weight of 104 tons, or 106,200 kilogrammes of boiler-room weights, and that a considerable addition has been made to these weights in the case of the later torpedo gunboats, in which locomotive boilers are used to develop the same power.

We are not in a position to give any information with regard to the space occupied by the eight water-tube boilers of the *Gaiser*, but, as we have shown above that the *Gaiser's* boilers were not worked to the uttermost, the matter is not of much importance as a basis for comparison; but we may say that in the steam yacht *Gitana*, belonging to the Baroness Rothschild, a very hard-worked locomotive boiler was replaced by a water-tube boiler with a very great advantage in the matter of stoking room, while the original length of the boiler room was reduced by two frame spaces, which were added to the cabin accommodation of the yacht. And in the case of a torpedo boat, two water-tube boilers were substituted for a single locomotive boiler which had given trouble, with no loss of stoking room, but with great advantage to the military efficiency of the vessel.

In conclusion, we beg to thank Mr. Ziese for the expression of

the esteem in which he holds Mr. Thornycroft, an expression which Mr. Thornycroft thoroughly appreciates.

We are, sir, your obedient servants,

JOHN I. THORNYCROFT AND CO.

CHURCH WHARF, CHISWICK, W.,

January 11, 1893.

NOTES.

HIGH SMOKE PIPES.

The following memorandum, submitted by the Engineer-in-Chief of the Navy to the Navy Department, is reprinted from the "Iron Age." It presents an interesting and valuable summary of the advantages to be derived from the use of high smoke pipes:

NAVY DEPARTMENT, BUREAU OF STEAM ENGINEERING,
WASHINGTON, October 14, 1892.

MEMORANDUM FOR THE HONORABLE THE SECRETARY OF THE NAVY.—ADVANTAGES OF THE 100-FOOT SMOKE PIPES OF *ARMORED CRUISER NO. 3.*

1. It enables a higher average speed to be steadily maintained without any special fittings.

A fair average value of the coal burned per square foot of grate surface per hour with an ordinary 60-foot pipe is 15 pounds.

It has been estimated, as the result of experience and experiment, that each 10 feet of additional height increase the draft as much as $\frac{1}{8}$ inch of water air pressure. Thus the extra 40 feet mean, in round numbers, $\frac{1}{2}$ inch of water air pressure.

A very conservative estimate of the increased combustion due to $\frac{1}{2}$ inch air pressure is 10 pounds per square foot. This makes the combustion with natural draft in the 100-foot pipe 25 pounds per square foot.

The effect of this in *Armored Cruiser No. 3*, with 1,016 square feet of grate surface is to give us in round numbers 11,500 indicated horse-power, instead of 7,000 indicated horse-power with the shorter pipe. As the cruising displacement is 9,100 tons, this means raising the sustained speed from 15.23 to 17.20 knots, nearly 2 knots per hour.

2. All experience goes to show that the long continued use of forced draft by pressure in the fire room has a deleterious effect on the boilers. Even as low a pressure as $\frac{1}{2}$ inch will, after a day or two, begin to develop leaky tubes.

This is avoided when the same combustion is produced by chimney draft.

3. Hence there is an increased longevity of the boilers when the increased power is secured by a high pipe rather than by forced draft, and a saving in the cost of repairs.

4. There is an actual saving at powers below the natural draft full power of all the boilers due to the following cause:

Whenever a boiler is under steam a certain amount of coal is expended simply in supplying the heat lost by radiation.

Experience with the smaller boilers of other ships leads to the belief that this amount for the large boilers for *Armored Cruiser No. 3* would be from $2\frac{1}{2}$ to $3\frac{1}{2}$ tons per day.

This means that where it would take (say) three boilers to give the desired speed with the short pipe two would do with the high pipe, making a saving due to laying off this boiler of at least 3 tons per day.

5. The high pipe will also produce an actual economy of combustion due to the more energetic combination of the oxygen of the air with the fuel. This has been shown by repeated experiments with moderate forced draft, to which, as has been seen, the action of the high pipe is equivalent in its good features.

6. There is also the saving of the extra coal needed to run the blowers. Reliable data of the power of blowers at various speeds are scarce, but available data suggest that the power of the eight large blowers of *Armored Cruiser No. 3*, to give $\frac{1}{2}$ inch air pressure would be between 50 and 75 indicated horse-power.

The simple engines driving these blowers are wasteful, and an allowance of 5 pounds of steam per indicated horse power is conservative. This means at the lowest figure taken (50 indicated horse-power) 250 pounds of coal extra per hour, or nearly 3 tons a day.

8. The necessary location of the blowers is such that it is impossible wholly to protect the working parts from dust, so that,

after steady running for some time, there is so much wear as to necessitate adjustment, or else to involve the risk of a breakdown. Such use, therefore, may prevent the machinery working at full power just when most needed.

It is thus seen that, besides the cost of steadily running the blowers in coal, there is the added cost of repairs.

8. The high smoke pipe gives us the benefit of an increased combustion due to ½ inch of water air pressure without requiring a single additional man to look after it, while the use of the blowers will involve all the time of one man in any case, and when all are working, several men.

This, moreover, comes at a time when extra work comes on the coal heavers and firemen, so that, with the ordinary peace complements, all the men are needed without allowing for some to look after the blowers. The subdivision of the coal bunkers makes the work of getting out coal arduous and slow at best, and, unless there is an adequate number, there is no use in trying to increase the rate of combustion, as the coal cannot be provided as fast as it can be burned.

9. There is the further advantage that, having already raised the steady cruising speed from 15.35 to 17.2 knots, if a small additional increase is wanted, say 1 knot, this can be obtained with a moderate increased air pressure from the blowers. With the short pipe ½ inch pressure would be required, besides the amount needed with the high pipe.

It is thus seen that the high pipe—

Increases the speed without injury to anything.

Increases the life of the boiler.

Increases the economy of combustion at moderate powers.

Saves coal lost by radiation from extra boilers used with short pipes.

Saves in repairs to boilers.

Saves in repairs to blowers.

Saves extra coal needed to run blowers.

It should, perhaps, have been distinctly stated at the start that this height of 100 feet is above the lowest grates of the boiler furnaces.

The clear height above boats and other large objects on the upper deck, which is the height that appeals to the eye, is only about 45 feet.

Respectfully submitted,

GEO. W. MELVILLE,
Engineer-in-Chief, U. S. Navy.

ON THE APPLICATIONS OF ELECTRICITY IN THE ROYAL DOCKYARDS AND NAVY.

BY MR. HENRY E. DEADMAN, CHIEF CONSTRUCTOR, PORTSMOUTH, ENGLAND, DOCKYARD.

[Extract from a paper read before the Institution of Mechanical Engineers, at the Summer Meeting, 1892, at Portsmouth.]

ELECTRIC COMMUNICATION.

The efficiency and safety of a modern warship are largely dependent upon the means of communication between the various working parts of the vessel. The officer directing from the conning bridge the movements of the vessel must have the means of instantly transmitting to the engine room a few prearranged signals with regard to the starting, stopping, etc., of the engines, and he likes to have the signal transmitted back to him to assure him that his order has been correctly received, and is about to be carried out. When manœuvring in squadrons, in which his ship has to keep her correct station in relation to the others, he must be able to communicate to the engine room his orders regarding small variations of speed, say one revolution more or less over a considerable range of speed. He also likes to be able to ascertain for himself on the bridge the actual revolutions of the engine; and he prefers to do this at a glance, without having to take the trouble to use a watch and count the revolutions say for a minute. Where the steering-wheel is immediately under his own eye, as it generally is when steam or other mechanical power is applied for steering the ship, he likes to have an indication on the bridge that the rudder is properly responding to the motions of the steering wheel. Should either of the alternative steering positions below have to be resorted to, he requires a certain means of communicating his orders to

the steersman at that position. In addition to the means of transmitting a few pre-arranged signals, he also requires a means of conversing from the conning bridge with the engine-room, with the steering positions, and with a few other places. In action, when the vessel's movements may be directed from a protected conning station, he needs, in addition to the foregoing requirements, to be put in communication with the officers at the various gun quarters, including turrets or barbettes, and also with those at torpedo-discharging positions, etc. In fact, he requires to be placed either directly or indirectly in communication with every part of the ship where officers and men are stationed. Where only a few pre-arranged signals are required, these inter-communications are effected by means of transmitting and receiving instruments, either mechanical or electrical; while for communications of a general character voice pipes are employed.

About six years ago, many complaints were received from ships as to the unsatisfactory nature of the means of signaling and intercommunication; and, as many new instruments, chiefly electrical, were then being proposed for use in the Navy, a committee was appointed, consisting of executive officers, naval constructors and naval engineers, of which the writer was a member, to inquire into and report upon the whole subject. In the course of the inquiry the views of a large number of representative naval officers were obtained, in order to determine, firstly, what intercommunication was necessary and best for a ship of war. On this primary question naval officers were not wholly agreed; but there was sufficient accord to enable a standard method to be laid down. In order to arrive at the best means for effecting these intercommunications, the committee studied all the methods that had been previously tried, including those in use in the mercantile marine. They carefully examined and tested many new instruments that were submitted to their notice, and carried out a number of independent experiments.

Voice Pipes and Call Bells.—Among other means of improvement, the committee recommended that all voice pipes connecting stations where there is noise or vibration, and all others, if

they are over 100 feet in length, should be increased from $1\frac{1}{4}$ inches internal diameter, the size then in use, to 2 inches internal diameter; and that, with these larger pipes, a system of electrical call-bells should be fitted. This plan is now generally carried out in the Navy, and has resulted in greatly improving the means of communication. The call-bell arrangement consists of an ordinary push board at the sending station, with the well-known annunciator or drop-shutter instrument at the receiving station; the dropping shutter indicates the station from which the call has come, and therefore the particular pipe to be spoken through. It is found that with voice pipes communication to places where there is much vibration or noise is not very satisfactory. Many devices have been and are still being tried with the object of improvement.

The committee were much impressed with the immense facilities offered by an electrical system of communication, and, amongst other things, telephones of different forms were tried in substitution for voice pipes. The results were not such as to lead to their introduction into the ships of the Navy.

Telegraphs.—With regard to instruments, though for ship work a mechanical system has some great advantages over an electrical, yet as a means of establishing communication between these instruments at distant stations the electrical system is far and away the best. The many disadvantages of a system of shafting and bevel wheels led through a considerable portion of the length of a ship scarcely need to be stated, while the great facilities offered by a simple insulated wire are equally self-evident. The committee, therefore, recommended that the one mechanical telegraph for communicating orders from the bridge or conning tower to the engine room should be retained, but that the telegraph for orders respecting speed of revolution should be electrical. Many of these electrical telegraphs have been fitted in the Navy, but the reports upon them have not been uniformly favorable, although in some instances they have been highly so. Whether the grounds for the unfavorable reports have always been good and sufficient is a matter of opinion, but at the present time electrical telegraphs in the Navy appear to be somewhat

under a cloud, so that the field is still open for invention in this direction. The telegraphs already in use are worked by primary or secondary batteries, while a new form has been proposed, to be worked direct from the ship's dynamo.

Testing of Dynamos.—Before acceptance from the makers all the dynamos purchased by the Admiralty are tested on receipt at a dockyard; and as no less than 150 have been tried at Portsmouth during the last two years, 1890 and 1891, the experience gained has been considerable. At the north end of the electric shop, a space is fitted with cast-iron holding-down plates, of sufficient area to take several sets of plant at one time; and also with an overhead traveler for lifting. Steam at 100 lbs. per square inch, or other pressure required, is supplied by three boilers of locomotive type in the adjoining boiler house. Adjacent to the holding-down bed is a separate enclosed room, containing all the various electrical testing instruments; and into this room the leads from the dynamos under test are taken. For enabling a dynamo on any part of the holding-down bed to be rapidly connected up with the instrument room, two bare copper rods extending overhead the whole length of the bed are fixed to the roof trusses and insulated by slate. From these are suspended two hanging leads, the lower ends of which can be connected to the terminals of any dynamo under trial. The overhead bars are coupled to leads, which pass through resistances sufficient to give the full load, and then into the instrument room.

Nearly all the dynamos now purchased, except those for torpedo boats, are of a capacity of 400 ampères at 80 volts, the latter being the Admiralty standard voltage for all dynamos and electrical fittings; and they are all direct-current and compound-wound. The dynamos are coupled direct to open vertical compound engines, working generally with 100 lbs. steam pressure at a speed of about 330 revolutions a minute. The dynamo and engine are required to stand first a continuous trial of six hours' duration with full load. Every half hour the steam pressure, current and electromotive force are recorded, as are also the temperatures of the testing room, field magnets and armature.

It is stipulated that one minute after the end of the trial the

temperature of any accessible part of the machine must not exceed that of the testing room by more than 30° Fahr, and the maximum temperature of the armature at end of trial after stopping must not exceed that of the room by more than 70° Fahr. If these limits are exceeded by more than 10° the machine is liable to rejection. These tests for temperature are considered to be of great importance, for if unsatisfactory the dynamo is not likely to remain long efficient. The spare armature which is supplied with each machine is required to undergo a similar test for two hours. The current produced by any dynamo is measured by means of a Siemens ammeter, and as a check upon this the current can also be switched on to a Siemens dynamometer. There are several of these ammeters and three dynamometers, of different sizes to suit the current to be measured. The leads pass through sliding resistance boxes, so that the strength of the current can be varied as required for the compounding trials. The voltmeter leads taken from the dynamo terminals are also led into the instrument room, and the electromotive force is ascertained by means of three Evershed marine voltmeters. A switch is interposed, of such a nature that either any one or any two or all three of these voltmeters may be in the circuit, thus checking one another's accuracy. A Cardew voltmeter and also a Siemens voltmeter can be switched in if desired. The Cardew voltmeter and the Siemens ammeter are the instruments actually used on board ships of the Navy. Very delicate instruments, such as electro-dynamometers, are found not to be suitable for use under such circumstances. As additional tests, the engines and dynamos are expected to stand without injury the sudden removal of the full load by the breaking of the circuit, and under such circumstances the maximum increase in revolutions or in voltage must not exceed 25 per cent. The dynamos are required also to be compounded, so as to give a constant electromotive force of 80 volts when the current is varied from 400 to 10 ampères, the speed being fairly constant during the time, and they are fully tested to ascertain that these conditions are fulfilled.

Engines.—As regards the engines, great importance is attached to their economy in the consumption of steam. The makers are required to state the consumption of water per electrical horse power per hour which they will guarantee shall not be exceeded. A fine is incurred for every pound of water per electrical horse power per hour in excess of the maximum guaranteed; and if the excess amounts to more than ten pounds per hour the machine is liable to rejection on this account. The water used is carefully measured, and every endeavor is made to avoid losses by leakage of steam, etc. Importance is also rightly attached to the governing arrangements, and it is stipulated that the increase of speed when the load is gradually removed must not exceed five per cent.

Resistances.—Before the six-hours' trial is started, and immediately after its conclusion, the resistances are observed of the dynamo complete, of the armature, and of the shunt and series windings of the field magnets; the resistances are thus obtained both when the circuits are cold and also when they are hot. For this purpose the leads from the terminals of the part whose resistance is required are taken into the testing room, and the resistances are obtained by a bridge made by the Silvertown Co., by a Thomson's marine galvanometer, and by an ordinary scale, for which the light is given by an incandescent lamp behind. The ordinary Daniell cells are used, and a key is introduced, so that one, two, three, or four cells may be switched in at once without altering the leads.

The dynamo and engine are further tried when fitted on board ship, and when the whole installation is complete.

ELECTRIC LIGHTING.

Messrs. J. H. Holmes & Co. have just fitted, in a most complete manner the S. S. *Tchihatchoff*, built by Messrs. Denny, of Dumbarton, for the Russian Steam Navigation Co., the installation having been carried out on the Holmes' improved single-wire system, and to meet the requirements of Lloyd's. On the top platform of the engine room two electric generating

plants are placed, each consisting of a combined engine and dynamo mounted on one bed plate, each dynamo being coupled direct to the crank shaft of its engine. The engines are of the open type, compound, each fitted with double cranks and automatic-expansion fly-wheel governor, acting directly on the slide valve. The dynamos are of the "Castle" type, and are each capable of supplying current for 200 sixteen-candle power lamps, when running at a speed of 200 revolutions per minute.

From the dynamo terminals, the main cables are taken to the main switchboard, which consists of an enameled slate slab encased in a polished mahogany frame, fitted with seven 50-ampère switches and fuses, also two ammeters, and one voltmeter. The main switchboard is so arranged that all lights can be run off one machine, or any particular circuit on either machine. Each switch is provided with an ivory tablet, on which the name of the circuit it controls is engraved.

Lead-covered cables are laid from the main switchboard to auxiliary switchboards, which are placed in convenient positions, with switches and fuses, each of which controls a branch circuit of eight or ten lights. No branches are taken from the main cables, except at the auxiliary boards. Distributing boards for saloon, stateroom and emigrants are fixed in first-class pantry, and each consists of an enameled slate slab, carrying the necessary switches and fuses. Each switch here has an ivory tablet fixed underneath, indicating the lights which it controls. These boards are fitted with neat mahogany frame and glass front, with lock and key. From these boards stranded conductors are led to the various groups of lights.

All wires are laid in wood casing to match the different woods throughout the ship, the covers being fastened with round-headed brass screws. The wires are insulated in a special manner with vulcanized india-rubber, then braided and compounded, the insulation resistance being 2,000 megohms per mile. Each coil is tested in water after being immersed at least twenty-four hours. The fittings are of brass, and of Messrs. Holmes & Co.'s own special design and make. Those in saloon and first-class accommodation are of the best Elkington electro-plate.

For the engine-room lights the cable is laid from the "engine-room" switch on the top platform. On the latter are placed switches with polished brass covers labeled respectively," top platform," "middle platform," "starting platform," "forward stokehold," "after stokehold," and "tunnel." From these switches, wires are laid to the various platforms, &c., as indicated by the tablets, branches being taken to the individual lights. By this method of distribution the lights are under the control of the engineer on watch, and he can light up any portion of the engine room and stokehold without leaving the electric engine space.

All the wires to the engine, stokehold, tunnel and gallery are, besides being insulated in the ordinary manner, further protected by a lead sheathing, and also a galvanized iron sheathing, and again braided over all. These are not cased in so as to be difficult of access, but fixed to the bulkhead by brass clips, and can therefore be easily traced and have a very neat appearance. No soldered joints are made in the engine room, all connections being made mechanically in cast-metal boxes, which are water-tight. By this means the system is reduced to its simplest form, and any mechanic can make a joint of this description. All fittings in the engine room are of Messrs. Holmes & Co.'s special design and make, every part being interchangeable.

There are four cargo lanterns, each containing eight 16-candle-power lamps, connected to a convenient length of armored flexible cable; also eight portable hand lamps for deck use in embarking or disembarking passengers. These lamps can also be used in the holds. The masthead and sidelights each contain a double filament of 32-candle power, and are controlled by switches placed in the wheelhouse. Altogether there are 270 lights installed throughout the ship.—"Marine Engineer."

THE REPAIRING OF THE *UMBRIA'S* SHAFT.

["Engineering," January 20, 1893.]

We are able this week to place on record an authentic narrative of the fracture and the subsequent repairing of the shaft of the Cunard liner *Umbria* during the protracted voyage which

occupied the last two weeks of the past year. The facts of the case could not be deduced from the narratives cabled to the daily newspapers, these being vague and contradictory. The vessel, as is well known, encountered very bad weather, strong gales and high seas, but all went well until the morning of December 23, five days out from Queenstown, when the thrust block rings began to move about a little. The engines were then slowed down and a careful watch kept. The steamer ran under these conditions for several hours, but about half-past five in the evening the chief engineer resolved to stop the machinery in order to have a thorough examination. This was commendable and timely, for it was discovered on lifting the thrust rings that the shaft was broken.

Figs. 1 to 3 show the thrust shaft of the *Umbria* as repaired, the fractures being marked on Fig. 1 by thick lines. The fracture extended along the neck close to a collar more than halfway round the circumference, then diagonally fore-and-aft to the next collar, and then back round the circumference for a short distance in the opposite direction. Adjoining each collar the cracks were very much opened.

Those who know the chief engineer of the *Umbria*, however, and the splendid resource which he has displayed under similarly trying difficulties, are not at all surprised that he at once determined to exeeute repairs and drive the steamer to New York, from which they were distant about a day's ordinary steaming. Coupling bolts being available, it was determined to proceed with the boring of three holes in each of the collars next the fracture, as shown on Figs. 1 and 3. This arangement helped to bind the shaft together endways, and put the torsional strain on the cross section of the bolts. The engineering staff in the ship worked hard and with great goodwill. They were divided into two watches of six hours, and for 72 hours the drills and chisels never stopped working. There is an old saying which has a wonderfully soothing influence even to such as are not always philosophic—"There was never a bad but there might have been a worse," and in this case it was brought home while operations were proceeding. Had the shaft broken in

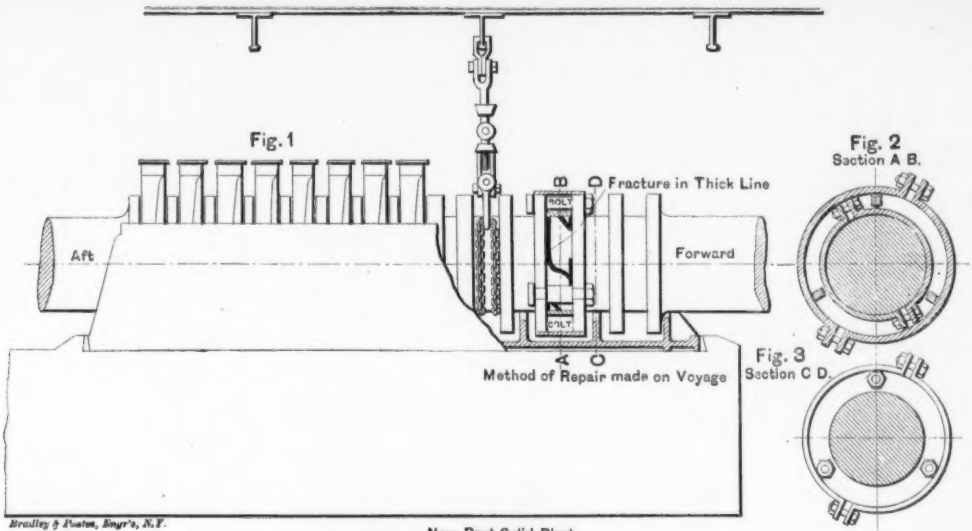
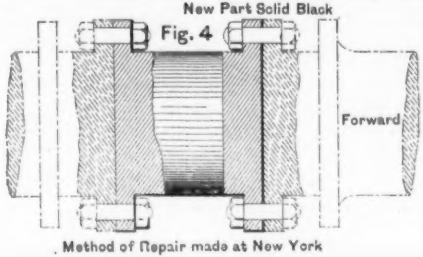


Fig. 1
Fracture in Thick Line
B
D
BOLT
BOLT
Aft
Forward
A
C
Method of Repair made on Voyage
Fig. 2
Section A B.
Fig. 3
Section C D.
Bradley & Foster, Engr's, N.Y.


New Part Solid Black
Fig. 4
Forward
Method of Repair made at New York

either of the adjoining spaces it would have been impossible to have repaired it in the way adopted, owing to the ribs which are cast on the base of the thrust block, and project upwards, as shown on Fig. 1. Even, as it was, the adjacent ribs gave clearance only for nuts two inches thick. A band was placed around the body of the shaft between the collars, and the bolts were then put into the holes. Around the neck and collars two straps were passed to secure the whole (Figs. 2 and 3). To insure that the shaft should not fall if anything went wrong with the clamp, the end of the shaft was hung to the deck beams by a chain sling and stretching screws, the shaft revolving in the bight of the sling (Fig. 1).

On the morning of the 27th ult. the work was finished, although the ordinary difficulties of working at tough steel in a close, confined space had not been lightened by the heavy sea which was running. A start was made with the engines working slowly, but two hours later a bolt broke. The weather was still rough, and it was considered prudent to stop the machinery again and replace the bolt. This took sixteen hours to accomplish, and when a start was again made the weather began to improve. The speed was gradually increased from 8½ knots to 10½ knots. The vessel steamed into New York in very fine weather.

To enable the ship to steam home, Mr. Tomlinson has designed the arrangement shown by Fig. 4, a piece of the shaft being cut out and a short length, having a flange at each end, being inserted and bolted to the collars of the thrust bearing, as shown. The chief engineer, Mr. L. Tomlinson, is to be congratulated on the successful completion of a heavy piece of work under trying conditions, and credit will readily be given by the profession for the watchfulness, pluck and resource shown on the occasion. The staff, it is acknowledged, displayed an *esprit de corps* which does great credit alike to the officer responsible for their organization and management, Mr. Bain, the superintending engineer of the company, and to the chief engineer of the vessel. Indeed, this spirit has ever been a characteristic of the staff of Cunard steamers, and might be traced to Sir John Burns, Bart., the chairman of the company.

BROKEN CRANK SHAFT OF S. S. *TRAVE*.

The following account of the way in which the *Trave* was worked into port with the low-pressure cylinder only, was contributed to the "Marine Engineer" by Mr. Leslie S. Robinson:

The mean speed of the *Trave* is usually about 17.75 knots, her main engine indicating 8,000 I.H.P. She was built by the Fairfield Shipbuilding and Engineering Co., in 1885 and 1886, for the North German Lloyds of Bremen.

On the voyage from Southampton to Bremerhaven, at 5 A.M., the engineer on watch heard a loud noise, and noticed that the L.P. crank remained stationary at short intervals, while the I.P. and H.P. cranks continued to revolve at a high speed. The engineer stopped the engine after she had made eight or nine complete revolutions. The chief engineer then came down and started the engines very slowly and cautiously. The I.P. and H.P. cranks made a quarter of a turn before the L.P. crank started, then they all made about half a turn together, and the L.P. crank stopped while the I.P. returned to its starting position. The angle between the I.P. and L.P. cranks was now 40° to 50° instead of 120°, and on raising the cap of the after I.P. main bearing, the crank shaft, which is 22 inches in diameter, was found to be fractured. The crank shaft is composed of Vickers' steel, and had been at work for about five years. It was thoroughly inspected about six weeks before the accident occurred and found to be free from flaws. It is a built-up crank shaft, the three cranks being interchangeable. The nature of the fracture leads to the conclusion that the cause of the accident was probably due to a flaw or blow hole in the material, as a portion of the fracture had the reddish appearance of oxidization due to salt water. But as the engine had made eight to ten complete revolutions since the accident occurred, the freshness of the fracture was somewhat destroyed. It now became necessary to anchor the vessel and prepare the engines to steam the remaining 188 miles to Bremerhaven with the L.P. engine alone if possible.

As the coupling bolts between the I.P. and L.P. cranks were jammed hard and could not be driven back, it was decided to

chip the fracture until there was sufficient clearance to run the L.P. engine. In the meantime the H.P. and I.P. valve casings were opened and the spring piston rings withdrawn from the valves, thus allowing high-pressure steam to pass direct through the H.P. and I.P. casings to the L.P. casing. The L.P. crank was placed a little over the top center, and at 2 P.M. steam was turned on and the engine made nearly a complete revolution. It could not complete it on account of the resistance at the fracture, there not being yet sufficient clearance. So they started to chip the shaft again, to give the necessary clearance. The chief engineer in the meantime packed up the L.P. valve nearly a quarter of an inch, to increase the bottom lead. He turned live steam on the condenser, started the circulating pumps, and put on the donkey to suck from the condenser. By this means he procured about 12 inches of vacuum.

At 4.30 P.M. the L.P. crank was again put just over the top center by the turning engine and steam was turned on, when the engine made four or five complete revolutions. The steam was then turned off in order to raise the anchor, and was turned on again immediately the anchor was aboard, and the vessel proceeded, without further interruption, to Bremerhaven at 42 to 47 revolutions per minute. The H.P. and I.P. engines began to revolve slowly, and at 9.30 P.M. they increased their speed, making about ten revolutions more than L.P. engine, which was doing all the work of propelling the ship.

At starting the boiler pressure was about 71 pounds; that of the H.P. casing, 57 pounds; the I.P., 28 pounds, and the L.P., 7 to 8½ pounds, the vacuum gauge standing at 26 inches.

As the boiler pressure rose, the stop valve was gradually closed, and the small valve for supplying steam to the I.P. casing was gradually opened. By 1 A.M. the boiler pressure had reached its normal, viz., 142 pounds, the main stop valve being almost shut, while the small one, steam to I.P. casing, was full open.

Under these conditions the H.P. and I.P. engines remained stationary. The pressure in the H.P. cylinder was 35 pounds; in the I.P., 14 pounds, and in the L.P., 8½ pounds.

In order to catch the high tide at Bremerhaven, which was at 7 A.M., the revolutions would have to be increased, and to attain this end the stop valve was again opened, whereupon the H.P. and I.P. engines again started to revolve at about 62 revolutions per minute, and the L.P. at 47 revolutions per minute.

At 8.45 A.M. the *Trave* dropped anchor before Bremerhaven, and the chief engineer tried the experiment of putting the engines in astern gear, which they responded to immediately.

From the above it is evident that the *Trave* had steamed 188 knots in 15 hours 25 minutes, or at the rate of 12.2 knots per hour, which, after allowing for the currents, gives a mean speed of about 13 knots per hour, or three-quarters of her nominal speed, and considering that she was steaming with her L.P. engine only, it is a most praiseworthy result.

The L.P. engine gave no cause for anxiety whatever during her fifteen hours' run, but the H.P. and I P. engines had to be most carefully watched and oiled. If the piston valves had been withdrawn from the H.P. casing, instead of the rings only, the two forward engines would certainly have remained stationary throughout the entire run.

CLASSIFIED SPEEDS OF BRITISH NAVAL VESSELS.

The Lords of the Admiralty have, in a revised regulation, authorized the following classification of the speeds of her Majesty's ships to be used on the different occasions referred to :

(*a*) The authorized power with natural draft = unit;

(*b*) "With all despatch" (the maximum power attainable for 24 hours) = $\frac{4}{5}$;

(*c*) "With despatch" (the maximum sea-going speed while the coal lasts) = $\frac{3}{5}$;

(*d*) "With moderate despatch" = $\frac{2}{5}$;

(*e*) "Ordinary speed" = $\frac{1}{5}$;

(*f*) "Most economical speed" = as determined by trial, and in accordance with the varying conditions of wind and weather.

The authorized power with natural draft (*a*) is to be used only during four hours of the quarterly passage trials or in an emergency; and the air pressures to be used for the attainment of

this power are not to exceed in the stokeholds half an inch of water for cylindrical boilers, and an inch of water for the locomotive boilers of torpedo gunboats, except when the boilers are fitted with automatic ashpit doors, in which case these air pressures may be exceeded by a quarter of an inch of water if necessary to obtain the power.

The $\frac{1}{3}$ power (*b*) is only to be used in cases of great urgency.

When steaming "with despatch" (*c*) all the boilers should be in use and the engine room staff in three watches; and when the coal is worked back, assistance should be given from the deck to bring the coal into more accessible positions if the work cannot be properly done without it.

"Ordinary speed" (*e*) is the speed at which passages are to be ordinarily made, but should be superseded by the "most economical speed" when it is required to steam the greatest distance for a given amount of coal. The "most economical speed" should also be used in those cases when the ordinary speed is less than the "most economical speed."

ALUMINUM.

["Engineering," December 23, 1892.]

As compared with most metals, pure aluminum, according to a recent article by Mr. A. E. Hunt, of the Pittsburgh Reduction Company, under ordinary circumstances, withstands the action of wind and weather exceedingly well; but the presence of silicon greatly reduces its resistance to atmospheric influences. Metal with 4 per cent. or 5 per cent. of silicon very soon collects a thick coating of oxide upon it, if severely exposed. Aluminum can be rolled or hammered cold, but the metal is most malleable at, and should be heated to, between 350 degrees and 400 degrees Fahr., for rolling or breaking down from the ingot to the best advantage. Like silver and gold, aluminum has to be frequently annealed, as it hardens up remarkably upon working. Due to this phenomenon of hardening during rolling, forging, stamping or drawing, the metal may be turned out very rigid in finished shape, so that it will answer excellently well for purposes where the annealed metal would be entirely too soft or too weak,

or lacking in rigidity, to answer. Especially is this true of aluminum alloyed with a small percentage of titanium, copper or silicon. It can be safely stated, as a general rule, that, under similar conditions, the purer the aluminum the softer and less rigid it is. Aluminum can be annealed by heating and allowing it to cool gradually; the best temperature is just below the red heat. Thin sections can be annealed by heating in boiling water. Aluminum can be easily and readily welded by electrical apparatus, and a cheap and satisfactory solder has been discovered. Sound castings of this metal can be made in dry sand molds or metal chills. It requires, however, some experience to master its peculiarities before sound castings can be uniformly made. The aluminum should not be heated very much beyond the melting point; if too hot it seems to absorb gases which remain in the metal, preventing sound castings.

SCHOLT AND GENS' NEW COMPOSITE GAUGE GLASSES.

["Engineering," December 9, 1892]

German science has already done much to improve the manufacture of glass for optical purposes, and it now appears that Dr. Scholt, of Jena, who has of late years done invaluable work in these improvements, has turned his attention in a more plebeian direction, and has succeeded in producing a gauge glass which will not crack from unequal cooling. In making ordinary gauge glasses, the glass is purposely cooled in the open air so as to cause the exterior of the glass, when cooled, to be in compression, so that, when heated and under internal pressure, the tension in the glass shall be reduced. This plan has, however, the disadvantage of leaving the interior of the glass in a state of severe tension. Dr. Scholt has, however, adopted the plan of making the gauge glass by combining two qualities of glass, having two different coefficients of expansion, by which he succeeded in greatly reducing their initial strains. Glasses made in this way have, in official tests at the German Imperial Laboratory, shown that they may be heated to 400° F., and will then stand without fracture drops of cold water being blown against the outside of the tube. In another test a Perth gauge glass

and one of Dr. Scholt's were both heated in oil up to a temperature of 420° to 430° F., and then plunged in cold water; the Perth glass cracked all over, whilst the new glass was as sound as before. These glasses have been tried with success on a number of locomotive and other high-pressure boilers upwards of 12 months, and have answered admirably. Messrs. Schaffer and Budenberg, of London, Manchester and Glasgow, are the agents for the United Kingdom.

LARGE PILLAR CRANE.

["Marine Engineer."]

The Clyde Trust's new mammoth 130-ton steam crane, now being erected on Stoffcross Quay, in Glasgow Harbor, is probably the largest pillar crane belonging to any Harbor Trust or Dock Co. in the United Kingdom, if not actually in the world. It was made by Messrs. Cowans, Sheldon & Co., Carlisle, in whose works it was tested under steam to the satisfaction of the engineers of the Clyde Trust. After erection it is to be tested up to 150 tons, for a working load of 130 tons, and the weights are to be raised with a steel wire rope to a height of 60 feet above the crane seat from a depth of 20 feet below the quay level—or a total lift of 100 feet. Its radius or sweep is to be 65 feet, and will project 45 feet beyond the face of the seat. The jib, formed of a double-steel tube, well-braced, is to be 90 feet long and 3 feet 3 inches diameter at the center of each tube, and its weight, including stays, 45 tons. The crane entire, in working order, will weigh 270 tons. The diameter of the rope drum will be 5 feet 2 inches, length 10 feet, and weight 10½ tons. The steel wire rope for 130 ton lifts will be 2⅛ inches diameter and 1,000 feet in length, and that for 20-ton lifts 1¼ inches diameter and 370 feet long. There will be two speeds for heavy lifts—2 feet per minute for 130 tons, and 4½ feet per minute for 60 tons—and a separate engine with two speeds for loads up to 20 tons—12 feet per minute for 20 tons and 30 feet per minute for 8 tons. For revolving there is to be another engine also of two speeds, one revolution in 6 minutes for 130 tons, and one in 3¼ minutes for 60 tons.

PORTLAND CEMENT FOR STEAM JOINTS.

["Engineering," December 9, 1892.]

To the Editor of "Engineering."

Sir: In your issue of November 25 I notice among the "Miscellanea" a note to the effect that a German inventor of Mayence has discovered that Portland cement may be used for joints more cheaply than other material now in vogue. I am afraid our Teutonic friend is a bit late with his "discovery." Seven years since, while serving as an engineer in the Manchester, Sheffield and Lincolnshire Railway Company's Continental steamboats, I saw it used for manhole doors. It was introduced into the company by the present superintendent, Mr. Mills, I think. Since then I have often used it at sea, not only for manhole doors, but ordinary steam and water joints. The way I used it was very simple, and not anything like the tedious and consequently expensive way you mention. If the cement is good, and a little common soda be mixed with it, all that is necessary is to cover the joint with it, nip up the bolts, and, thirty minutes after, the pressure, either steam or water, may be safely admitted to the pipe. I have many times seen the manhole doors put on and steam raised within from four to six hours after, and long before that the cement would be set perfectly hard. In fact, if the cement did not show signs of setting within a few minutes after applying it, it would be rejected and some fresh made. Yours truly,

WALTER J. WOOD,
Superintendent Engineer International Steam Trawling Company.

EXPERIMENTAL TANK FOR RUSSIA.

["Marine Engineer."]

For conducting speed experiments with ship's models in a specially constructed and equipped tank, such as the late Dr. William Froude established at Torquay a number of years ago, and such as his talented son, Mr. R. E. Froude, still directs at Gosport, and Messrs. William Denny & Bros. have established at their works in Dumbarton, Messrs. Kelso & Co., model mak-

ers and electricians, of 2 Commerce street, Glasgow, are supplying the Russian naval authorities with the necessary and very special outfit in the way of dynamometric apparatus in use at these establishments. This outfit has just been finished, and will shortly be dispatched to St. Petersburg, and there fitted up in a specially constructed tank in connection with the arsenal.

It comprises the dynamometer "model truck" or carriage, to which the model to be experimented with is harnessed while being towed through the tank, together with all the electric and other gear for recording, in graphic form, speed, resistance, time, distance, &c., a most elaborate and skilfully designed and finished piece of mechanism. This carriage is designed to hold the model so that it will be towed at a uniform speed along the whole length of tank at the same time that it is subject to the resistance and buoyant motion due to the fluid in which it freely floats, and through which it passes.

The outfit also includes the "screw truck" or carriage which carries the screw propeller, or propellers, proposed to be fitted to any given vessel, and which can be moved and adjusted to give the propellers any desired distance—longitudinal or transverse—from the stern post of the vessel.

A third and equally intricate piece of the entire outfit is a forerunner, or current meter, whose object is to record the state of the water in front of the model and screw trucks, so that when experiments have to be taken frequently at short intervals, the influence due to the currents in the water may be allowed for. Messrs. Kelso & Co. who were also the makers of the apparatus in use by Messrs. Denny & Bros., at their private tank in Dumbarton, supplied, about three years ago, apparatus of the same kind to the Italian Government, and it is now in operation at the naval works at Spezzia. In each case, through the courtesy of our own Admiralty and Mr. R. E. Froude, Messrs. Kelso & Co. and the commissioners representing the foreign governments who have adopted the system have been permitted to avail themselves of and reproduce the latest improvements which the experience of Mr. Froude and the experts assisting have suggested.

BASIC BESSEMER STEEL.

The Navy Department has rescinded Circular No. 43 (published on page 112 of Volume IV of the JOURNAL), which permitted the use of basic Bessemer steel wherever open-hearth steel was specified.

No case has arisen where any contractor desired to take advantage of this permission, so that it may be considered that the uniform practice in the building of our new ships and machinery has been to use open-hearth steel.

THE PECK-WHEELER SYSTEM OF HIGH TEMPERATURE (COMBINED LIVE AND EXHAUST STEAM) FEED-WATER HEATING.

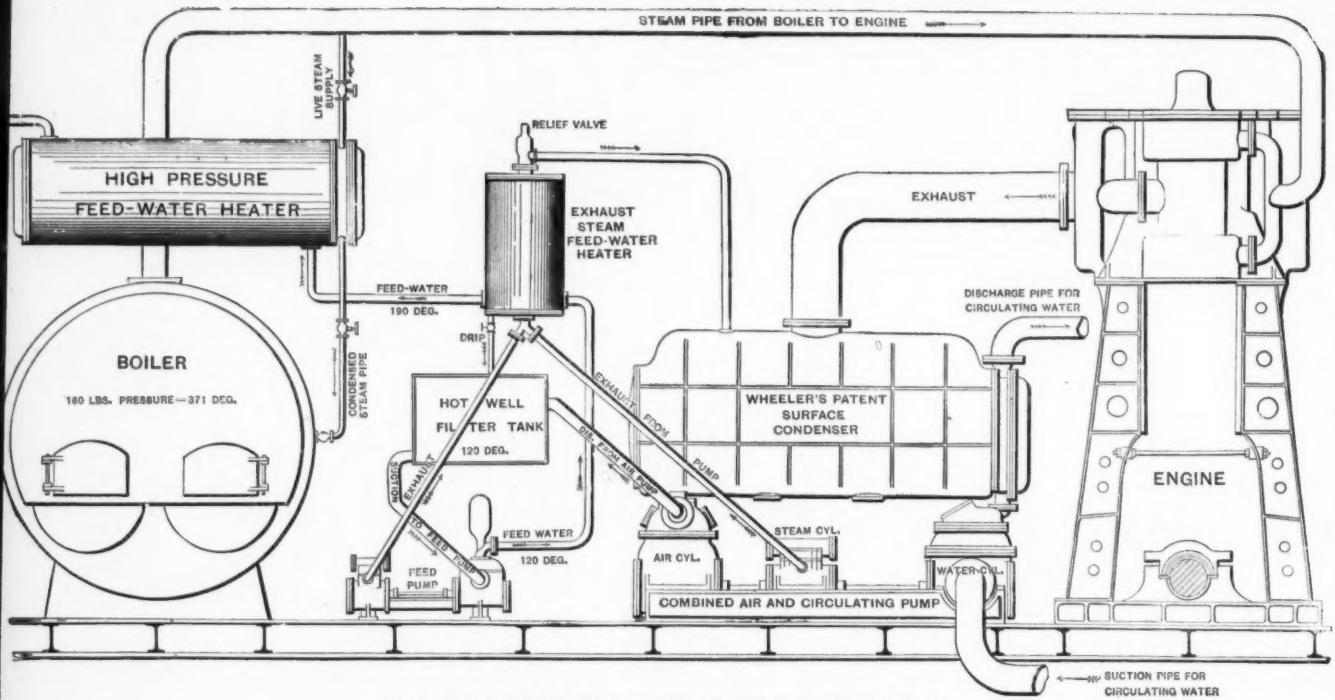
Supplying steam boilers with feed water at a very high temperature (at or near the temperature of the steam in the boiler) is a subject worthy of the attention which it is now receiving in this country and abroad.

The advantages of this system will become apparent on an investigation of the subject, and, although it would seem an impossibility, facts prove that to heat feed water by the use of "live steam" taken direct from the boiler is productive of economy.

The higher the feed water is in temperature, the greater the economy in fuel and the longer the life of the boiler, to say nothing of the considerable increase of its steaming capacity.

Heating feed water by steam taken directly from the boiler, although tried many years ago, was never followed out to any successful issue (except for purifying feed water) until recently.

The Peck-Wheeler system represents the successful result of exhaustive experimenting. Its object is, first, to obtain all the heat possible out of the exhaust steam from the main engine by working so as to get the temperature of the feed water in the hotwell as high as is consistent with a good working vacuum—say a temperature of 120° F.; second, to further heat the feed water by passing it through a small exhaust steam heater, into which the feed pump, air pump and circulating pump, and any other steam pump or auxiliary engine may exhaust, such exhaust steam being entirely condensed by the feed water; at the same time, the latter is raised to a temperature of from 180° to 190°



THE PECK-WHEELER SYSTEM OF HIGH TEMPERATURE
COMBINED LIVE AND EXHAUST STEAM FEED-WATER HEATING, PATENTED.

F. as it passes on its way to the "live steam" heater. The feed water, after being heated to maximum temperature in this special "live steam" heater, enters the boiler at very nearly the temperature of the steam within the boiler. The steam, which is condensed in producing this result, flows to the boiler by gravitation. The annexed illustration shows clearly the operation of the apparatus. The steam pressure in the boiler is, say, 160 pounds gauge pressure, representing about 371° F. On the main steam pipe, near the boiler, is shown a branch pipe for supplying "live steam" to the high pressure feed-water heater. After the steam is used in the triple-expansion engine, it passes through the exhaust pipe to the surface condenser, as shown.

The water of condensation is taken by the air pump (which is immediately under the condenser), and discharged into the hot-well filter tank, where the temperature averages 120 degrees F. This feed water, after passing through the filtering material in the tank, flows to the feed pump, which forces the water through the small exhaust steam feed-water heater, where (as before stated) it is raised in temperature to about 180 to 190 degres F. The cut shows the exhaust steam from the combined air and circulating pump and the feed pump exhausting into said heater, where it is entirely condensed by the feed water passing through its coils or tubes, the water of condensation gravitating to the hot-well filter tank below and mingling with the feed water therein.

The feed water, after passing through the small supplemental exhaust heater, enters the bottom of the "live-steam" heater at one end, and flows out at top at the other end, where it emerges at a temperature of about 355 degrees F., and then passes down to the boiler. The live steam is taken from the main steam pipe, as shown, or from the boiler direct, as preferred. This steam passes through the inside of the tubes of the heater, and after being condensed gravitates to the boiler through the vertical pipe shown on the other side of the boiler.

The advantages claimed for the "Peck-Wheeler" system are as follows:

1st. Economy in the use of fuel, varying from 6 to 10 per cent.

THE PECK-WHEELER SYSTEM OF HIGH TEMPERATURE COMBINED LIVE AND EXHAUST STEAM FEED-WATER HEATING, PATENTED.

2d. Increase of steaming capacity of the boiler, varying from 15 to 20 per cent.

3d. Longer life to the boiler, as the plates and tubes are not exposed to the severe strains due to feed water of a low temperature.

4th. Purifying the feed water in the heater.

5th. Less fluctuation of steam pressure in the boiler—in other words, steam generation is more continuous and prompt.

6th. Less weight, space and first cost for boiler, and less expense for repairs.

The reason why the boiler is more free from injury with hot feed water is stated by the designer of the apparatus under description, as follows:

When water averaging only 120 degrees F. is pumped into a boiler carrying, say, 160 pounds steam pressure, it cannot boil until heated to 371 degrees F., or 251 degrees above the temperature at which it was fed into the boiler. When thus pumped into the boiler this feed water of low temperature gradually mingles with the water already there, reducing the temperature of the latter below the boiling point, and checking evaporation until the resulting mixture of boiling water and feed water is again heated to the boiling point. The water in the boiler is then expanded by the heat, becoming lighter, so that its particles move more freely among themselves, and thus create a more rapid circulation. As the temperature rises, the internal currents, by which the heat is carried off from the heating surface, become more rapid, and consequently more heat is carried away from said surfaces. In other words, after the water commences to boil, the buoyant steam bubbles, rising through the mass of water, greatly assist in increasing the circulation, thus causing the heat to be absorbed from the heating surfaces with the greatest rapidity. It will thus be seen that it is desirable to keep the whole body of water in the boiler in a continuous state of ebullition by supplying feed water at the highest possible temperature in order to attain and maintain the most effective conditions, since heat can be transmitted to the water at about three times the rapidity while evaporation in the form of boiling is going on,

as compared with the same water while being heated to the boiling point.

A naval engineer,* recently writing on this subject, speaks of a marine double-ended boiler twenty feet long where the difference of expansion between the two opposite parts of the shell of the boiler would amount to about one-third of an inch with the old system of feeding water at low temperature. With feed water approximating "live steam" temperature, such strains are entirely avoided. He also adds that the tubes of the "live steam" heater serve to a large extent as a grease extractor, and also for arresting scale-making properties in the feed water. Ready means, of course, should be provided for cleaning out the heater. He also cites the case of an English vessel that employed a "live steam" heater whose use increased the revolutions of the engine and the steam pressure of the boiler, and also made the work of the fireman less laborious.

The "London Engineer," from which the example was taken, goes on to say:

"The facts ascertained were simply that the heater gave a higher boiler pressure by five pounds, and raised the revolutions of the engines from 34 to 35½ per minute. As regards economy we have no data to go on save that the firemen let the furnaces more alone with the heater than without."

"It remains to be considered how this remarkable result can be reconciled with thermo-dynamic laws. At first sight it appears to be impossible that any result of the kind could be got, or that any economy should result. If we take into account the well-known fact that heating surface for some reason not fully understood, is more efficient with boiling water than with cold; that all grease is kept out of the boiler, being deposited in the heater, from which it is periodically removed with soda, and that so the surfaces are kept cleaner, we can at once see that a saving of five or six per cent. is not more than is to be expected. The 'live steam' heater being supplied with steam from a boiler working with feed water at a very high temperature indeed, the

* Assistant Engineer T. W. Kinkaid, U. S. N., in his article entitled "Feed-water Heating," in the August, 1892, number of the JOURNAL.

efficiency of each pound of fuel employed to heat that feed will be greater than it would be if the steam were supplied by a boiler working with cold feed.

"The fact that there is a greater gain than calculation indicates as possible is explained by the more or less uncertainty which exists as to the influence of the density of the water in a boiler on the rate at which it will take up heat. Each unit, again, of heating surface will probably, in a given time, transmit about the same quantity of heat, whether the water in contact with it is 200 degrees or 350 degrees, or even hotter, because the furnace temperature is always so much in excess of the water temperature that 100 degrees or 150 degrees one way or the other makes little difference; but it is obvious that, if the water be very hot, more steam will be produced in a given time per unit of surface than if it is cold. The result is equivalent to an augmentation of boiler heating surface, and, as in the case referred to, more steam will be made per hour than could be had from the same boilers with cold water. The steam taken for heating the feed is used to the best possible advantage, the whole of its contained heat being utilized; whereas, if the same steam were passed through the engine, not more than 12 per cent. of its heat could be utilized by the engine."

The illustration and description have been kindly supplied by the Wheeler Condenser and Engineering Company, 92 Liberty street, New York.

SHIPS.

UNITED STATES.

Brooklyn (Armored Cruiser No. 3).—Bids for the construction of this vessel were opened at the Navy Department, December 15, 1892, with the following result:

On the plans of the Navy Department (described in Vol. IV, p. 805, of JOURNAL):

Newport News Shipbuilding and Dry Dock Company,	$3,147,000
Union Iron Works, of San Francico,	3,050,000
Cramp & Sons,	2,986,000

On plans submitted by bidder:

Bath Iron Works, of Bath, Me.,	$3,165,000
Cramp & Sons (*a*),	3,086,000
Cramp & Sons (*b*),	2,880,000

Bid *a* of the Messrs. Cramp was for the Department's hull with quadruple-expansion engines of their own design, the other parts of the machinery being in keeping with this change. The boiler pressure was to be 200 pounds, and forced draft was to be on the "induced" system.

Bid *b* was for a hull similar to that of the *New York* with quadruple-expansion engines, and the other parts of the machinery as in bid *a*.

The idea in both cases was to use the engine as quadruple expansion at full power, and, for low powers, to disconnect the low-pressure connecting rod and valve gear and use the remaining three cylinders as a triple-expansion engine. It was claimed that this plan was better than the two sets of engines in the Department's plans, as it avoided the unequal wear of the shafts of the forward and after sets of engines.

The plans were very carefully considered by the Bureaus of

Construction and Repair and of Steam Engineering, which recommended jointly that the Department's plans be adhered to.

This recommendation was adopted by the Department, and the bid of the Messrs. Cramp being the lowest on the Department's plan, the contract was awarded to them.

Iowa (Sea-Going Battleship No. 1).—Bids for the construciton of this vessel were opened at the Navy Department December 15, 1892, with the following result:

On the plans of the Navy Department (described in Vol. IV, p. 811, of JOURNAL):

Newport News Shipbuilding and Dry Dock Company,	$3,233,000
Union Iron Works of San Francisco,	3,150,000
Bath Iron Works of Bath, Maine,	3,185,000
Cramp & Sons,	3,010,000

On plans submitted by bidder:

Cramp & Sons (*a*),	$3,110,000
Cramp & Sons (*b*,)	2,870,000

In the case of this vessel, as in that of the *Armored Cruiser*, the bids on their own designs by Messrs. Cramp were for quadruple-expansion engines worked at 200 pounds pressure, induced draft, and the other changes incident to this modification. Bid *a* contemplated the Department's hull, while bid *b* took the hull of the *Coast-line Battleships*.

As in the case of the *Armored Cruiser*, the plans were referred to the Bureaus, which, after careful consideration, recommended adherence to the Department's plans.

The contract was awarded to the Messrs. Cramp on their bid of $3,010,000.

Katahdin (Harbor Defense Ram) was launched at the Bath Iron Works on February 4th. A description has already appeared in the JOURNAL (Vol. II, p. 576), but is here repeated. Length, extreme, 243 feet; length on water line, 242 feet 9 inches; breadth, 43 feet 5 inches; breadth at water line, 41 feet 10 inches; draught of water, 15 feet; displacement, about 2,050 tons. The thickness of armor of conning tower is 18 inches; thickness of

deck armor at side, 6 inches; thickness of deck armor at center, 2 inches; thickness of armor of smoke pipe and ventilators is 6 inches.

The vessel is constructed with a double bottom, the entire length of the ship, 2 feet between the skins, and divided into numerous separate cells. The framing is on the longitudinal system, and great strength is given to the structure by the longitudinals and girders being continuous from stem to stern. The depth of the longitudinals throughout their length is 24 inches, and the girders which support the armor deck are 15 inches in depth.

The transverse frames within the double bottom are intercostal brackets. The vertical keel, four of the longitudinal frames and the armor shelves are to be water tight; also the transverse frames, about every twenty feet, are to be water tight; thereby the space between the outer and inner shells is divided into about 70 water-tight compartments. The vessel above the inner skin is divided by transverse and longitudinal bulkheads into 30 water-tight compartments, making in all about 100 water-tight compartments. All hatches through the armor deck are to have battle plates of thick armor.

The vessel is to be lighted by electricity and is to be submerged to the fighting trim by means of Kingston valves in the different water-tight compartments.

Artificial ventilation is to be on the exhaust system. The vitiated air is to be withdrawn by fans; the fresh air to be supplied through ducts connected to the armored ventilators.

The propelling engines will be placed in water tight compartments, and separated by an athwartship bulkhead. They will be of the horizontal, direct-acting, triple-expansion type, each with cylinders 25, 36 and 56 inches diameter by 36 inches stroke. Indicated horse power about 4,800, at 150 revolutions per minute. The main valves will be of the piston type, worked by Marshall valve gear. There will be one piston valve for each high-pressure and each intermediate cylinder, and two for each low-pressure cylinder.

The bed plates and cylinders will be secured together by forged steel tie-rods, and supported on wrought steel keelson plates built in the vessel. All crank, line, thrust and propeller shafting will be hollow. The shafts, piston rods, connecting rods and working parts generally will be of forged steel.

The condensers will be made of composition and sheet brass. The air and circulating pumps will be independent. The propellers will be of manganese bronze or equivalent metal.

Each engine room will have an auxiliary condenser with combined air and circulating pump.

There will be two double-ended boilers, 13 feet 8 inches diameter by 22 feet 6 inches long, and one single-ended boiler, 13 feet 8 inches diameter by 11 feet 7 inches long, for a working pressure of 160 pounds per square inch. They will be placed in two water-tight compartments, and will have one smokepipe.

Independent boiler feed pumps will be placed in the fire rooms.

The forced-draft system will consist of two blowers for each fire room, which will be air-tight. There will be steam reversing gear, ash hoist, turning engines, auxiliary pumps, engine room ventilating fans, distilling apparatus, engineers' workshops, etc.

Bancroft (Naval Academy Practice Vessel).—This little vessel completed a most successful trial on January 26th, maintaining an average speed for four hours of 14.374 knots. As the horse power had not been computed in time to make the account of the trial complete, we shall postpone an extended notice until the May number of the JOURNAL. Meanwhile, the following account of the details of determining the speed is given.

Special interest attaches to the trial of the *Bancroft* because it is the first case in which the speed was determined by the method proposed by Engineer-in-Chief Melville and unanimously approved by the Board on Construction. This method has already been described in the JOURNAL (Vol. II, p. 73). Briefly stated, it consists in standardizing the screws by a series of progressive trials over the measured mile so as to determine accurately the correspondence of speed and revolutions. (The mean

of the revolutions of the two shafts is used.) A curve is constructed exhibiting the relation, and the vessel is then taken to sea for a straightaway run of four hours. During this run, the revolutions, as shown by the engine counters, are recorded every fifteen minutes. At the end of the four hours, the total number of revolutions made by each engine is determined, the mean taken and divided by 240, thus giving the average revolutions per minute. By applying this to the curve of speed and revolutions, the speed is at once determined.

In the contracts for these speed trials, there is a provision for premiums and penalties amounting, in some cases, to $2,000 for each hundredth of a knot. It follows, therefore, that in using this method some means, more accurate than the ordinary engine counter, had to be provided for determining the time and revolutions *on the measured mile.* This led to the construction of the Weaver Recording Apparatus, which is a specially designed chronograph and revolution recorder. It is described in the JOURNAL (Vol. III, p. 86). This enables the time to be determined to hundredths of a second and the revolutions to hundredths.

Since the *Bancroft's* trial, articles in the press have appeared intimating that it was an error to attribute the method to Commodore Melville, and claiming that the same method had already been used, notably in the trials of the *Cushing.* These statements disclose a misapprehension of Commodore Melville's plan, which it may be well to dissipate.

It would be hard for anyone, nowadays, to claim originality for determining the speed of a vessel by the revolutions of the propellers, as this has been done for years, and is, in fact, the method of "dead reckoning" employed in many steamers. It is, however, only an approximation, although, when done carefully, a close one. It is likewise true that the *Cushing's* speed was determined by first standardizing the screw on the measured mile, although it is not known whether this was the first case or not. In her case, however, the standardization was to determine the number of revolutions required to give the contract speed,

and, so long as the number was maintained on the contract trial, nothing more was required. There was no speed premium or penalty, and it was not necessary to determine the exact speed so long as the contract speed was maintained.

The vital point in Commodore Melville's method, and the only one for which originality can be claimed, is that it makes provision for determining exactly what the speed on the contract trial is to the hundredth of a knot. This it does by combining the progressive measured-mile trials with the record of revolutions, both of which, of course, are entirely familiar.

It is apparent that there is no difficulty if the contract speed is less than the highest obtained on the progressive trials, but it might be thought that no provision has been made for the possibility of exceeding this highest previously determined speed. This point, however, was anticipated and is provided for. To any one who is familiar with the method of laying out these curves of performance, it is well known that the natural position taken by the "spline" or batten in laying out the curve at the highest observed point is identical for a reasonable distance with the curve that would be drawn if a higher point had been observed. Consequently, in laying down the curve of speed and revolutions, it is prolonged by following the natural position of the spline or batten for about a knot, so that, if the contract revolutions exceed those obtained on the measured mile, the exact speed can still be accurately and exactly determined from the curve.

Just here it may be well to call attention to several points in which this method is superior to other accurate methods of determining the speed.

First, is its convenience. No more observers are required than if an ordinary set of progressive trials was to be made, and no course has to be laid off, as the accurate measured mile with ample depth of water has existed at Newport since 1889. When the *Philadelphia* was tried, a forty-mile course had to be laid off, and half a dozen ships were stationed along it to secure current observations, while a pilot ship had to precede her on

the course to see that the vessels were at their stations and that the course was clear.

Second. The result of the trial is known within ten minutes of its termination, and there are no corrections or changes to be made. A long course trial involves the correction for current, which, of necessity, must be made afterwards, and may involve considerable delay.

Third. The speed can be determined at any time by simply counting the revolutions, so that the contractors know exactly where they stand.

Fourth. The record of the counters every fifteen minutes enables the contractors to extend the trial and then take any continuous four-hour period, in case it is found that the performance improves as the trial proceeds. This is impossible on a long course, where it would be necessary to run over the entire course again.

Fifth. Fog need not interfere with the trial in the slightest degree if it is not made on a frequented course. The only observations for speed are made on the engine counters and these are not affected. A fog of any intensity would prevent a long-course trial.

Sixth. Although not of any bearing on the contract trial, it is an additional advantage that the standardized screw method gives the data of accurate progressive trials, thereby putting the designers in possession of data of the greatest value for subsequent designs.

Seventh. The speed and revolution curve will be of great assistance to the navigating officers in the subsequent regular cruising in furnishing the means of accurately determining the dead reckoning instead of having to depend on the erratic patent log.

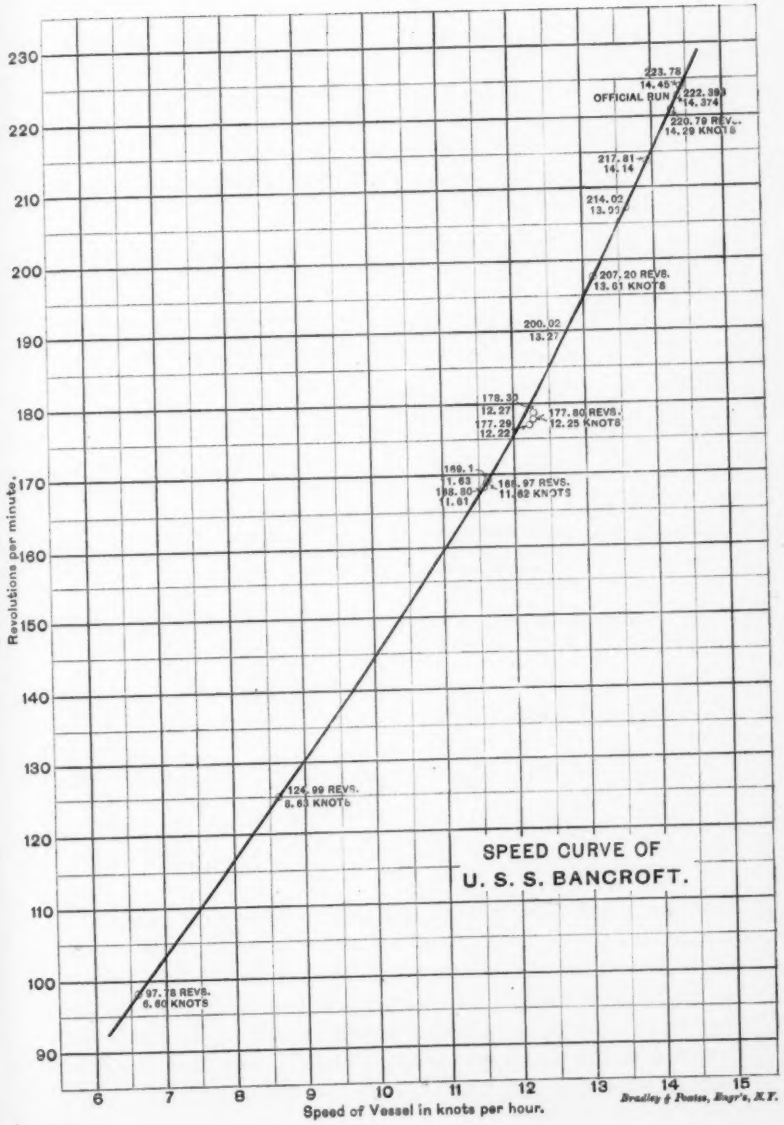
The *Bancroft's* progressive trials were made over the measured mile at Newport on January 21st and 23d, consisting of two runs (one each way) at 7 and 9 knots, and four (two each way) at 11.6, 12.25, 13.6 and 14.3 knots. The data for these runs are given in the accompanying table:

Table I.

RECORD ON MEASURED MILE BY WEAVER APPARATUS.

No. of run.	Time on the mile.	Revolutions on the mile.			Revs. per min. (mean both shafts.)	Mean speed knots.	Second Means.	
		Starboard.	Port.	Mean of both shafts.			Speed knots.	Revs. per min.
	min. sec.							
Up.	10 13·69	1,006.24	1,010.62					
1 Down.	7 56 90	767.86	769 94					
Mean.	9 05·30	887.06	890.29	888.67	97.78	6.60	6.60	97.78
Up.	7 12·23	905 32	885.90					
2 Down.	6 41·59	852.79	829 87					
Mean.	6 56·91	879.06	857.89	868.47	124.99	8 63	8.63	124.99
Up.	5 14·17	873.84	877.56					
3 Down.	5 05·88	857.26	879.96					
Mean.	5 10·02	865.55	878.76	872.16	168 80	11.61		
Up.	5 10·92	863.25	871.63				11.62	168.97
4 Down.	5 07·88	872.15	882.23					
Mean.	5 09·40	867.70	876.93	872.32	169.15	11 63		
Down.	5 02·95	900.08	886.99					
5 Up.	4 46·36	847.54	848.25					
Mean.	4 54·66	873.81	867.62	870.72	177.29	12.22		
Down.	5 03·91	898.92	897 85				12.25	177.80
6 Up.	4 42·85	845.81	844 92					
Mean.	4 53·38	872.37	871.39	871.88	178.30	12.27		
Down.	4 35 03	927.53	928.40					
7 Up.	4 27·68	882.92	879.46					
Mean.	4 31·35	905.22	503 93	904.57	200.02	13.27		
Down.	4 20·99	932.64	929.93				13.61	207.02
8 Up.	4 14·84	910.73	906.62					
Mean.	4 17 91	921.69	918.28	919.98	214.02	13 96		
Down.	4 12·33	916.22	928.12					
9 Up.	4 16·90	926 54	926.25					
Mean.	4 14·61	921.38	927.19	924.28	217.81	14.14		
Down.	4 05·76	912.90	912.64				14.29	220.79
10 Up.	4 12·66	946.47	945.41					
Mean.	4 09·21	929.69	929.02	929.36	223.78	14.45		

From these data the curve of speed and revolutions was plotted on a large scale, which would enable hundredths of a knot to be measured accurately. This curve, on a reduced scale, is annexed.


SPEED CURVE OF
U. S. S. BANCROFT.
Revolutions per minute.
Speed of Vessel in knots per hour.
97.78 REVS.
6.80 KNOTS
124.99 REVS.
8.63 KNOTS
169.1
11.63
168.80
11.61
165.97 REVS.
11.62 KNOTS
178.30
12.27
177.29
12.22
177.80 REVS.
12.25 KNOTS
200.02
13.27
207.20 REVS.
13.61 KNOTS
214.02
13.93
217.81
14.14
223.78
14.45
OFFICIAL RUN
222.393
14.374
220.79 REVS.
14.29 KNOTS
Bradley & Pontus, Engr's, N.Y.

The contract trial was made on January 26th, and consisted of a four-hours' straightaway run almost south from Beaver Tail Light. As already stated, the mean speed was 14.374 knots, corresponding to a mean number of revolutions of 222.393.

The following table gives the avarage revolutions per minute for each fifteen-minute period.

Table II.

REVOLUTIONS FOR FIFTEEN-MINUTE PERIODS ON FOUR-HOUR OFFICIAL TRIAL.

	Average revolutions per minute for the fifteen minutes ending at this time—			Remarks.
	Starboard.	Port.	Mean.	
A. M. 8·45	228.066	225.00	226.53	
9·00	226.60	224 80	225.70	Counter readings at 8.45 A. M.:
9·15	224.33	226.33	225.33	Starboard, 224809;
9·30	223.27	224 07	223.67	Port, . . 224156.
9·45	220.13	222.27	221 20	Counter readings at 12.45 P. M.:
10 00	224.20	223.00	223 60	Starboard, 278352;
10·15	222.47	221.93	222.20	Port, . . 277362.
10·30	222.06	222.10	222.09	Total revolutions in four hours:
10·45	221·93	221.33	221.63	Starboard, 53543
11·00	219.60	219.67	219.63	Port, . . . 53206
11·15	224.20	221.07	222.63	
11·30	225.27	219 80	222.53	Mean, . 53374·5
11·45 P. M.	225.30	220.80	223.05	Average revolutions per minute for the trial:
12·00	225.07	221.27	223.17	$\frac{53374\cdot5}{240} = 222.393.$
12·15	221.27	218.67	219 97	
12·30	221.70	219.67	220.68	
12 45	222.14	220.27	221.20	
Means.	223.09	221.69	222 392	

During the contract trial two patent taffrail logs were used, each with 400 feet of line. Both were new, but the one referred to as *B*, in the following table, had been used on the trip of the *Bancroft*, from the contractor's works at Elizabethport, N. J., to Newport. It had underlogged the vessel as shown by comparison with bearings. These logs were not tested on the measured mile, but on the way out of Narragansett Bay, before the contract

trial, their readings were taken for a distance known by bearings to be about two knots. On this distance log A was correct, while log B was 0.2 knot short.

The readings for the four-hour trial were as follows:

	Log A.	Log B.
1st hour	14.7	13.5
2d "	14.7	13.6
3d "	14.6	13.5
4th "	14.7	13.5
Average	14.675	13.525

The trial was a marked success in every way. The steam pressure and revolutions were maintained with marked uniformity, and the machinery gave entire satisfaction, not a drop of water being used on any of the bearings. There was scarcely any vibration, a glass of water, evenly full, standing on the cabin deck without spilling.

The water at the bow rose about a foot with a flat top and no spray. At the stern there was very little disturbance.

The speed called for in the contract was 12 knots, the boilers being worked under an air pressure of one-half inch of water.

In explanation of the remarkable excess shown on the trial above the contract requirements, it may be said that, when the *Bancroft* was designed, it was expected that the air-pressure would be about an inch, as in the case of the *Yorktown*, and all the details of the design were based on this. After the design had been made, the air pressure was reduced, and it became necessary to fix upon a new contract speed. As no data were available of accurate trials with half an inch air pressure, an estimate was based on the *Yorktown's* performance with an inch. It was supposed that about twelve knots and a half could be obtained, if everything worked perfectly, and, in accordance with the Department's practice of allowing a slight margin for unforeseen contingencies, twelve knots was fixed as the contract speed.

The *Bancroft* has a remarkably fine hull, the lines being the design, it is understood, of the late Naval Constructor Gatewood.

This accounts for a part of the superiority over the *Yorktown's* record. In addition, the contractors used special machinery, with which they have been building high-speed stationary engines, for making all working parts absolutely true, the piston and valve rods and shafting being all "ground true." Selected Pocahontas coal was also used, while the *Yorktown* had anthracite. Finally, the *Bancroft's* engines are vertical, while those of the *Yorktown* were horizontal.

The splendid performance of the machinery and the marked success of the trial reflects the highest credit on the builders, the Samuel L. Moore & Sons Company, of Elizabethport, New Jersey.

ENGLAND.

Pique, Rainbow and Retribution.—Messrs. Palmer's Shipbuilding and Iron Company, Limited, Jarrow-on-Tyne, have delivered to the Admiralty the three cruisers which they were intrusted to build under the Naval Defence Act. The vessels are second-class cruisers, of the following leading dimensions and particulars:

Length between perpendiculars,	300 feet.
Breadth, extreme,	43 feet 8 inches.
Depth, molded,	22 feet 9 inches.
Displacement on a mean draught of 17 feet 6 inches,	3,600 tons.
Indicated horse power,	9,000.
Speed,	19¾ knots.

Each vessel has two funnels and two pole masts, with a light fore-and-aft rig. The hull throughout is built of steel, and the bottom of the ship is sheathed with teak, 3½ inches in thickness to 2 feet above the water line. The stem, sternpost, rudder and shaft brackets are of phosphor-bronze. There is a steel protective deck which extends the whole length of the vessel. The transverse section of this deck is in the form of a flat arch, the crown of which rises 1 foot above the water line at the center of the vessel, and slopes down towards the sides to a point 4 feet below the load line. The thickness is 2 inches on the sloping

part and 1 inch on the crown. The engines and boilers, magazines, steering gear, &c., and other "vitals" of the ship are placed under the protective deck. Vertical engines are adopted, and in order to insure protection to the cylinders, which project above the protective deck, a belt of armor of solid steel 5 inches in thickness with 7 inches of teak backing is fitted round the engine hatchways between the protective and upper decks. Each vessel is divided into a large number of water-tight compartments, and has a double bottom the full extent of the engine and boiler space, the continuity of which is carried forward and aft by the water-tight flats forming the magazines, shell rooms and store rooms. The bunkers are placed alongside the engines and boilers, and extend to the upper deck. There are two separate engine and boiler rooms. Above the protective deck aft are the cabins for the officers, the part amidships being occupied by coal bunkers, engineers' workshops, washplaces, &c., while the forward part is entirely devoted to the crew. Under the poop are placed the cabins of the commander and principal officers, ward rooms, &c., the forecastle being taken up by the crew. The armament of each vessel consists of two 6-inch breechloading guns, one mounted on the poop and the other on the forecastle, both placed on the middle line of the ship; six 4.7-inch quick-firing guns, three on each broadside; eight 6-pounder quick-firing guns, one 3-pounder gun, and four 5-barrel Nordenfelt guns, mounted on suitable stations along the sides of the vessel; a 9-pounder gun for boat and field purposes is also secured on deck. In addition to this, four torpedo tubes are fitted, one forward, one aft, and one on each broadside. For controlling the ship in action a conning tower of steel 3 inches thick is fitted on the aft end of forecastle, inside of which the various telegraph instruments, steering wheel, voice pipes, &c., are placed. As regards the pumping arrangements, an elaborate system of piping is fitted, extending to every compartment, while the fire service and sanitary arrangements are on the same elaborate scale. A complete installation of electric light is also fitted, including three powerful search lights. The crew numbers 250 hands all told.

The engines are vertical, of triple-expansion type, and drive twin screws. They were designed and constructed by the Palmer's Company under the direction of Mr. J. W. Reed, manager of the engine works. Each set of engines is placed in a separate water-tight compartment, these being divided by a middle-line bulkhead. The diameters of the cylinders are 33½ inches, 49 inches and 74 inches, all with 39 inches stroke of piston. They are placed with the high-pressure cylinder forward, the intermediate pressure in the middle and the low pressure aft. The valve chests are placed on the forward sides of the high-pressure and intermediate cylinders and on the after side of the low-pressure cylinders, a piston valve being fitted to the high pressure, while the intermediate pressure and low pressure have double-ported flat slide valves, fitted with Church's relief rings in the back. The cylinders are quite independent castings, but connected by bar stays and copper pipes. The breeches standards are also connected near their tops by struts. The liners of the high pressure and intermediate cylinders are of steel, while that of the low pressure is of cast iron. The pistons, together with the cylinder and valve-chest covers, are all of cast steel. This latter material is largely used in the framework of the engines. The back standards are of the split type of box section, and of cast steel with the piston-rod guides attached. The bed plate for each set is in three independent pieces, each cylinder having its own bed plate. These are of box form and of cast steel, and both bed plate and standards are of a very substantial character. Each cylinder is supported at the front by two turned columns of forged steel, an additional column being placed under the low-pressure valve chest. A diagonal stay runs from the bottom of each front column up to the under side of the cylinders at the back, and the engines are further stiffened by an athwartship stay being carried from each cylinder top to a horizontal fore-and-aft girder formed on the middle-line bulkhead. This makes an unusually stiff framework, and the intention of the designers was fully realized, for on the full-power trials, when running 145 revolutions, there was not the slightest vibration even at the tops of the cylinders. The crank shaft is of steel, and in three pieces, with the cranks placed

at 120 degrees apart. Both shaft and pins are hollow, the diameter of the shaft being 12¾ inches. The piston and connecting rods, together with most of the gear, are of steel.

The valves are driven by ordinary link motion, with the reversing shaft placed along the back of the engines. The reversing engine has two cylinders, and is snugly placed on the front of the intermediate cylinder. The reversing gear is of the ordinary all-round worm-and-wheel type. The handwheel in front may be put out of gear when not required. The various handles and wheels are grouped together conveniently at the front. There is one single-acting air pump placed at the back of each low-pressure engine, driven by levers from low-pressure crosshead. The condenser is of cylindrical form, of cast gun-metal, and placed at the after end of the engines. The diameter of the body is 6 feet 6 inches, and the length 8 feet. The total cooling surface for both condensers is 10,040 square feet, with tubes ⅝ inch in external diameter. The circulating water for each condenser is supplied by a 14-inch Tangye centrifugal pump. There is a cross connection between these two engine rooms, so that one pump may supply both condensers in case of accident to the other. Each pump is capable of discharging 750 tons of water per hour from the bilges. The main feed pumps are of the Weir's duplex type, one pair being being placed in each engine room. The speed of the pump is controlled by a float in the feed tank. A double cylinder pump of Admiralty pattern is placed in each engine room for fire and bilge purposes. In each engine room there are also a Normandy's evaporator and condenser, air-compressing engines and reservoirs, by Belliss, and electric light engine and dynamo by Willans and Robinson. A separate auxiliary condenser, provided with its own air and circulating pumps, is fitted in each engine room for condensing the steam used by all the auxiliary engines throughout the ship. The combined surface of the two condensers is 1,000 square feet. The positions of the various auxiliary engines have been carefully chosen, and the general arrangement of these and the main engines is such that there is unusually ample room to get easily and comfortably about.

The propeller shafting is of steel throughout, and hollow, the external diameters being 12¾ inches and 14½ inches. The propellers are of Admiralty gun metal, with three blades on each. They are about 13 feet in diameter and 17 feet 6 inches in pitch.

There are five return-tube boilers, occupying two separate watertight compartments. Three of the boilers are double-ended, 13 feet in diameter by 18 feet 6 inches long, and two are single-ended, 13 feet in diameter by 9 feet 6 inches long, the working pressure in all cases being 155 pounds per square inch. There are in all twenty-four corrugated furnaces, each having a separate combustion chamber. The total heating suraface is 15,704 square feet, and the total fire-grate surface 588 square feet. The boilers are arranged for working under forced draft on the closed stokehold system, the air being supplied by eight double-breasted fans. In the boiler rooms there are placed for auxiliary purposes, three double-cylinder feed pumps of Admiralty pattern, one of them being especially for one of the single-ended boilers, which is intended to be used for auxiliary purposes when in harbor. Provision is made in the double bottom under the boilers for the storage of a large supply of fresh water, and connections are made from the auxiliary feed pumps to these tanks. The steam from the after boilers and from the forward boilers is led to the engine rooms by two independent lines of pipes, so that in the event of one of the boiler rooms being flooded, or its steam pipe damaged, the steam can be taken from the remaining boiler room to both sets of engines. The main steam pipes are lapped with copper wire as an additional security. In addition to the various engines already mentioned, there are ash hoists, drain-tank engines, steering engines and capstan engines. A workshop is also fitted up with several machines, and a special engine to drive them.

The builders undertook to develop 7,000 indicated horse power, with an air pressure in the stokehold not more than one-half inch, for eight consecutive hours, and 9,000 indicated horse power with not more than 1¼ inches, for four consecutive hours. All of the vessels have been successfully tried. The trials were made off Plymouth.

The *Pique* went on her trials on January 27 and 30, 1892, the result of the eight hours' trial showing 7,515 indicated horse power with .28-inch air pressure, and on the four-hours' trial 9,258 indicated horse power with 1.05 inches air pressure, the results in both instances being considerably better than required by contract, and the working of the machinery and boilers very satisfactory.

The *Rainbow* went on her trials on May 13 and 18, 1892. On the eight-hours' trial 7,879 indicated horse power was developed

H. M. S. *RAINBOW* EIGHT-HOURS' TRIAL, OFF PLYMOUTH, MAY 13, 1892.

Hours.	Steam in boiler room.	Revolutions.		Vacuum.		I.H.P.		Collective I.H.P.
		Star-board.	Port.	Star-board.	Port.	Star-board.	Port.	
1	143	129.4	131.3	29	28	3,864	3,997	7,861
2	143	130.5	130.5	29	28	3,911	3,934	7,845
3	143	132.2	132.9	29	28	3,968	4,042	8,010
4	145	131.1	131.9	29	28	3,917	3,923	7,840
5	150	132.9	134.5	29	28	4,066	4,095	8,161
6	144	131	133	29	28	4,007	4,000	8,007
7	148	131.3	132.4	29	28	3,856	3,932	7,788
8	148	128	129.5	29	28	3,726	3,796	7,522
Mean.....	145	130.8	132	29	28	3,914	3,965	7,879

Mean air pressure .32 in. (dampers closed frequently).

FOUR-HOURS' FORCED-DRAFT TRIAL, OFF PLYMOUTH, MAY 18, 1892.

Half hours.	Steam in boiler room.	Revolutions.		Vacuum.		I.H.P.		Collective I.H.P.
		Star-board.	Port.	Star-board.	Port.	Star-board.	Port.	
1	150	138.6	141.1	28.5	27.5	4.609	4,960	9,569
2	153	142.1	142.5	28.5	27.5	4.912	5,136	10,048
3	150	140.2	140	28	27.5	4.610	4,677	9,287
4	150	141.5	142.2	28.2	27.2	4,851	4,925	9,776
5	152	141.6	143	28.3	27.3	4.999	4,913	9,912
6	151	142.1	142.1	28.2	27.2	4.979	4,919	9,898
7	150	140.6	141.5	28.3	27.3	4,919	4,741	9,660
8	151	141.5	141.8	28.2	27.2	4,916	4,867	9 783
Mean.....	151	141	141.8	28.3	27.3	4,849	4,892	9,741

Mean air pressure, .69 in.

with .32 inch of air, and on the four-hours' trial 9,741 indicated horse power with .69-inch air pressure. In the eight-hours' trials of both vessels, even with the low air pressure, it was frequently necessary to close the ash-pit dampers to check the excessive generation of steam, the fans being run as slowly as possible without stopping. Appended is a detailed statement of the results of *Rainbow's* trials.

H. M. S. *Retribution*, the third and last cruiser, underwent her trials on September 16 and 19. On the former day she went outside Plymouth breakwater and had a most successful eight-hours' trial, everything working in a most satisfactory manner. The contract power was exceeded by 645 indicated horse power, although the air pressure was only one-half that allowed by contract. The mean results were :

Steam in boilers,	144 pounds.
Steam in engine room,	141 pounds.
Air pressure,	0.25 inches.
Revolutions,	128.8
Vacuum,	28.1 inches.
Collective indicated horse-power,	7,645

On the 19th she went out again for her four-hours' trial under forced draft, and this trial also was marked with the same success as attended the natural-draft trial. There was nothing to be gained by exceeding the contract power to any extent, and no attempt was made to show a high indicated horse power, as the *Rainbow* had shown clearly what could be done. The object in the present case was only to obtain the contract power with a reasonable margin. The result showed that the contract power was exceeded by 367 with an air pressure of less than half that allowed by contract. The mean results were :

Steam in boilers,	148 pounds.
Steam in engine room,	145 pounds.
Air pressure,	.61 inches.
Revolutions,	138.8
Vaccuum,	28 inches.
Collective indicated horse power,	9,367

Throughout both the trials the engines worked remarkably well, and the boilers gave a very free supply of steam. No signs of leaky tubes developed during the trials, and at the conclusion the boilers were very tight and free from leakage.

The comparative ease with which the majority of the cruisers of this class develop their contract power has clearly shown the wisdom of providing an ample amount of boiler power. The approximate speed of the *Rainbow* by log, on the eight-hours' trial, was 19.4 knots, and on the four-hours' trial 20.2 knots per hour. The contractors, however, have no guarantee of speed.

Sappho and Scylla.—These vessels are second-class cruisers, built under the Naval Defence Act, from the designs of W. H. White, Esq., C.B., Director of Naval Construction.

They were built by Messrs. Samuda Bros., of Poplar. The principal dimensions of the vessels are: Length between perpendiculars, 300 feet; breadth, extreme, 43 feet; depth, 23 feet 3 inches; displacement, 3,400 tons; and they are equipped with two 6-inch breech-loading guns, six 4.7-inch quick-firing guns, and nine 3-pounder quick-firing guns.

The engines are of the triple-expansion type, vertical, twin screw. They were constructed by Messrs. John Penn & Sons. The diameters of the cylinders are 33½ inches, 49 inches and 74 inches, all of 39 inches stroke. The H.P. cylinders are placed forward, and are fitted with piston valves, the I.P. and L.P. valves being of the double-ported flat-valve type. The liners are made of steel for the H.P. and I.P. cylinders, and of cast iron for the L.P. cylinders. The pistons, cylinder covers and steam-chest doors are of cast steel. The cylinders are carried on round forged-steel columns on the front side and cast-steel ones at the back, these latter carrying the piston-rod guides. The bed plates are also of cast steel of box section, and are strongly secured to bearers built in the ship. To secure lightness, the main connecting rods are made hollow.

The crank shaft is hollow and made of steel with the cranks at 120°.

The condensers are made entirely of brass, the total cooling surface being 10,000 square feet. The circulating water is sup-

plied by 14-inch centrifugal pumps made by Messsrs. Tangyes, of Birmingham. These pumps are arranged to pump out the bilge in the event of leak, at the rate of 750 tons per hour. The reversing gear is of the ordinary link motion type with double bar links and adjustable working parts. Steam reversing engine and hand gear are fitted.

The air pump is entirely of brass, and is worked from the L.P. piston rod crosshead. The main feed pumps are by Messrs. J. & G. Weir, of Glasgow, the feed pipes being $3\frac{1}{2}$-inch diameter. A fire and bilge pump of Admiralty pattern is fitted in each engine room.

A Normandy distiller in combination with a Weir evaporator is furnished for each engine room, as also air compressors and reservoirs by Belliss & Co., and electric light engines by Thornycroft & Co., with dynamos by Siemens & Co.

An auxiliary condenser, made of brass, is fitted in each engine room for condensing the steam of all the auxiliary engines on board.

The propellers are of gun metal, three bladed, and 13 feet diameter.

The propeller shafting is made of hollow steel, $12\frac{3}{4}$ inches diameter inboard, and $14\frac{1}{2}$ inches outside of the ship.

The boilers are of the return-tube type of Siemens-Martin steel throughout, three of them being double-ended, 13 feet diameter by 18 feet 6 inches long, and the other two of the same diameter and 9 feet 6 inches long, the working pressure being 155 pounds per square inch.

The furnaces are Fox's corrugated, 24 in number, with a separate combustion chamber to each furnace. The total heating surface is 15,770 square feet, and the grate 590 square feet. Forced-draft fans are fitted, delivering into closed stokeholds. These fans are eight in number, 5 feet diameter, made by Peter Brotherhood, and fitted with his 3-cylinder engines. The main steam pipes are wound with copper wire for extra safety.

The trials of the *Sappho* were in all respects highly satisfactory, the I.H.P. under natural draft being 7,301 mean of eight-hours' trial, and the maximum I.H.P. 7,689. The four-hours'

forced-draft trial resulted in a mean of 9,617 I.H.P., and a maximum of 9,861 I.H.P. The ship traveled at 19.41 knots at natural draft, and 20.47 knots at forced draft.

The official trials of the *Scylla* took place off Sheerness during the week ending December 17th. On Thursday, the 15th, the vessel underwent her natural-draft trial, when the mean I.H.P. developed was 7,622, with an air pressure in the stokeholds of .2 inch, and on the 17th inst. the vessel went through her forced-draft trials, when the mean I.H.P. realized was 9,280.5, giving a speed of 20.625 knots. Both trials were most successfully carried out. The following is a summary of the forced-draft trial:

Draught of water forward, 13 feet 3 inches; aft, 17 feet 5 inches. Speed of ship, 20.625 by log. Steam pressure in boilers, 145.08. Air pressure in stokeholds, .87 inch. Vacuum in condensers, starboard, 27.4; port, 27.5. Revolutions per minute, starboard, 139.7; port, 137.5. Mean pressure in cylinders, starboard, high, 52.9; intermediate, 30.03; low, 15.6; port, high, 55.4; intermediate, 28.9; low, 15.31. I.H.P., starboard, 4,696.68; port, 4,583.84; collectively, 9,280.52.

Hercules.—The third-class battleship *Hercules* returned into Portsmouth Harbor on the 23d of November, after having tried her engines and guns in the Channel. During her former commissions she was propelled with a two-bladed screw, but after she was equipped with new triple-expansion engines of the modern type, she was fitted with a four-bladed propeller having a diameter of 22 feet 6 inches and a mean pitch of 17 feet 6 inches. So far as power and speed were concerned, this proved eminently satisfactory, but the vibration that was set up, owing, as it was supposed, to the increased number of revolutions and the want of sufficient clearance between the blades and the run of the ship was so excessive that it threatened to tear the hull to pieces. A portion was subsequently cut off from each blade, reducing the diameter to 21 feet 6 inches, but when tried under weigh, there was found to be no appreciable difference in the violence and noise attending the vibration. The *Hercules* was again docked and the boss relieved of two of the blades, the pitch at

the same time being fined to 16 feet 6 inches. The trial of November 22d was conducted under these conditions. As it was not deemed expedient to deliver the same thrust upon the two blades as upon the four, the engines were worked at half power, which was obtained under the altered circumstances of reduced resistance with the same number of revolutions (81) that had previously secured the full power of 7,500 horses. There was a perceptible decrease in the vibration, and when the engines were driven astern, it ceased altogether, thereby showing that all that was wanted was additional space for the escape of the water thrown up by the screw. It is now proposed to fit the ship with a stronger two-bladed propeller having a coarser pitch, and to set the blades further to the rear in order to secure increased clearance.

Royal Oak.—On Saturday, November 5th, Messrs. Laird Bros., of Birkenhead, floated from the building dock the *Royal Oak*. This is one of the eight first-class battleships ordered under the Naval Defense Act of 1889, four being built in H. M. dockyards and four by contract in private yards. They will be the largest and most formidable ships ever constructed for the British Navy. With the exception of one, these ships will be of the barbette type.

The dimensions of the *Royal Oak* are: Length, 380 feet; beam, 75 feet; displacement, 14,300 tons; and the mean draught as designed is 27 feet 6 inches. Her machinery, which has been constructed by Messrs. Laird, consists of two sets of triple-expansion engines with cylinders 40 inches, 59 inches and 88 inches diameter, and 4 feet 3 inches stroke, working at a pressure of 150 pounds and driving twin screws which are of gun-metal. These have four blades and are 17 feet diameter.

The boilers, which are now complete and ready to lift on board, are eight in number, single-ended and built of steel; each has four corrugated furnaces, the two center ones with common, and the wing with separate, combustion chambers. The weight of the boilers is about 36 tons each.

The machinery is designed to develop 9,000 I.H.P. under natural draft, and 13,000 I.H.P. under forced draft, and the esti-

mated speed of the ship is 16 knots and 17½ knots under these conditions respectively. The hull is built entirely of mild steel, and is constructed of great strength; the stem, stern post and stern brackets for carrying the propeller shafts are steel castings. The vessel has a complete double bottom, and, with several fore-and-aft, and many athwartship, bulkheads, is most efficiently subdivided. The greatest thickness of armor on the broadside is 18 inches, and on the barbettes 17 inches. The protective steel deck extends below the water line from the bow for about 76 feet, and from the stern for about 72 feet; and on an armor shelf at the level of this deck is built a belt of steel-faced armor supported by a backing of teak, which extends from 5 feet 6 inches below the load water line to about 3 feet above it. The total length of the armored belt is about 250 feet, and there are armored bulkheads across the ship at its ends. The sides of the ship above the thick armor are protected by a lighter belt of 4-inch plates, behind which are the coal bunkers, which materially add to the protective power by what is known as coal protection; on the top of the belt armor a 3-inch protective deck is worked across the ship so that armored-deck protection is afforded for the whole length of the vessel.

The armament will consist of four 13½-inch 67-ton guns mounted in pairs in the barbettes, and firing a projectile weighing 1,250 pounds, with a powder charge of 630 pounds; ten 6-inch 100-pounder quick-firing guns—four on the main deck in armor casemates 6 inches thick, supplied by Messrs. Vickers, Sons & Co., which are completely fitted in place, and six on the upper deck, the two forward and the two after ones mounted on sponsons, allowing of a right ahead and astern fire; sixteen 6-pounder and nine 3-pounder quick-firing guns, and eight small machine guns, and two 9-pounder field guns. The auxiliary armament is well distributed from bow to stern, and some of the smaller pieces are mounted on the bridges and in the military tops. The vessel is also fitted with seven torpedo tubes, five being above water and two being submerged, and she will carry on board 18 torpedoes. The total weight of the armament is about 1,900 tons.

The guns and torpedoes will be fired by electricity, and for the lighting of the vessel, which will be fitted with over 650 lamps and four electric search lights of 25,000 candle power, a very powerful apparatus is being fitted.

The first plate of the keel of the *Royal Oak* was lowered into the dock on the 29th May, 1890, and up to the present time nearly 9,000 tons of material of all kinds, some in form of complicated and highly finished machinery, and all involving a large amount of labor per ton, have been worked into her.

Revenge.—The first-class battleship *Revenge*, one of the largest battleships hitherto constructed, was launched on November 3d, 1892. It is the scond of the two building by Palmer's Shipbuilding and Iron Co., Limited, the first one, H. M. S. *Resolution*, being launched on the 28th May last. These vessels are of the barbette type.

The dimensions and particulars of the *Revenge* are as follows: Length, 380 feet; breadth, 75 feet; draught, mean, 27 feet 6 inches; displacement, 14,150 tons; freeboard, forward, 19 feet 6 inches; freeboard, aft, 18 feet; I.H.P., forced draft, 13,000; I.H.P., natural draft, 9,000; speed, forced draft, 17½ knots; speed, natural draft, 16 knots. The coal carried at the designed load draught is 900 tons. The construction of the ship has been made exceptionally strong. The hull alone absorbs not less than 9,640 tons of the total displacement, and of that amount about 7,200 tons has already been worked into the structure. She is built entirely of mild steel, the stem, sternpost, rudder and shaft brackets being formed of cast steel.

The hull, being divided into 220 water-tight compartments, reduces to the fullest extent the risk of danger to bottom plating from rocks and torpedoes, and renders it practically unsinkable. There is a double bottom extending throughout the engine room, boiler and main magazine spaces. The inner bottom is raised at the center of the ship, and forms the flat for the magazines, which extend from the inner bottom to the lower deck. The engines and boilers are separated by longitudinal bulkheads extending the whole length of the magazine space. Longitudinal bulkheads at the sides extend throughout the machinery

space, and form coal bunkers and wing spaces. On the platform, debris and lower decks, is placed the auxiliary machinery for the working of the ship, including steering engines, electric engines and hydraulic pumping engines, as well as a fully-equipped workshop and numerous store rooms.

The upper deck extends from stem to stern, without a break, and above it are the shelter decks, on which are the conning towers, two in number; these are surmounted by flying bridges connected with each other by a fore-and-aft bridge. The boats, of which there are twenty-one, including two 56-feet topedo boats, are stowed amidships. A strong steel derrick is fitted to to the main mast for lifting them, and the foremast is also fitted with a derrick for working those of a lighter description.

The masts, which are built of steel, are fitted with military and signaling tops, and there are two funnels. The barbettes project through the upper deck a few feet, and inside are powerful hydraulic turning engines, and all the gear for controlling the turntables and also for working the guns.

A sloping protective deck of steel $2\frac{1}{2}$ inches in thickness extends under water from the bow for about 76 feet, and from the stern for a distance of about 72 feet, and between these two points there is a protective deck 3 inches in thickness, worked horizontally about 3 feet above the water line; from the level of this deck there is a steel-faced armored belt 18 inches in thickness and 8 feet 6 inches wide, extending for a distance of 250 feet of the midship part of the vessel, the thickness tapering at the ends to 14 inches. Immediately above this belt there is also a light belt of armor 4 inches thick, extending for a distance of 144 feet, and terminating in bulkheads at each end 3 inches thick, which extend from side of ship to sides of barbettes; behind this thin belt of armor, coal bunkers are arranged whereby a large amount of additional protection is secured above the thick armor belt.

The four 6-inch guns on main deck are placed in casemates, protected by armor plates 6 inches thick. At the forward and after ends of the belt, and rising directly from the protective deck, are the barbettes formed of steel-faced armor, 17 inches

thick. With a view of preventing water from finding its way below the protective deck, means are provided for closing the several openings by watertight covers, while, in the case of those which must necessarily remain open, cofferdams have been fitted with the same object.

The main armament consists of four 67-ton breech-loading guns of 13½-inch caliber, with a training of 120 degrees on each side of the center line. The auxiliary armament consists of the following: ten 6-inch 100-pounder quick-firing guns, four in armored casemates on the main deck, and six on the upper deck; sixteen 6-pounder quick-firing guns, four being on upper deck and twelve on main deck; nine 3-pounder quick-firing guns, three in military tops, and six for boats; two 9-pounder field guns, eight Nordenfelt guns, and seven torpedo tubes, four on the broadside, one at the stern, and two submerged. The total weight of the main armament is 1,410 tons, and the weight of the auxliary armament is 500 tons.

The *Revenge* will be lighted throughout by electricity with an installation of about 700 electric lamps, and will also be equipped with four electric search lights of 25,000 candle power, each of which will be worked by a dynamo under protection. Means are so arranged that the ship, when in action, for fighting, can do so from either of the two conning towers. The thickness of the forward conning tower is 14 inches, and the after one 13 inches.

The *Revenge* is a twin screw, each set of engines having cylinders 40 inches, 59 inches, 88 inches, by 51 inches stroke. There are eight single-ended boilers, each 15 feet 6 inches diameter by 9 feet 6 inches long, having in all 32 furnaces, working at a pressure of 155 pounds. There are no less than 69 auxiliary engines, *i. e.*, including steering engine, electric light engines and dynamos, air compressing, distilling, evaporator, boat-hoisting, and workshop engines.

The 900 tons of coal carried at the designed load line will enable her to steam 5,000 knots at a 10-knot speed, but in case of necessity she will be able to stow about 400 tons more, and so obtain a radius of action of over 7,000 knots. When used as

a flag ship the *Revenge* will have a complement of over 700 officers and men.

DETAILS OF BRITISH WARSHIPS LAUNCHED 1892.

Name.	Displacement.	I.H.P.	Cost.	Builders.	Engines.
	Tons.		£		
Repulse	14,150	13,000	849,553	Pembroke.	Humphreys.
Ramillies	14,150	13,000	877 460	Thomson.	Thomson.
Revenge	14,150	13,000	852,755	Palmer.	Palmer.
Resolution	14,150	13,000	852 755	Palmer.	Palmer.
Royal Oak	14,150	13,000	877,378	Laird.	Laird.
Centurion	10,500	13,000	608,098	Portsmouth.	Scott, Greenock.
Barfleur	10,500	13,000	599,089	Chatham.	ditto.
Crescent	7,700	12,000	377,474	Portsmouth.	Penn.
St George	7,700	12,000	373,258	Earle Co.	Maudsley.
Gibraltar	7,700	12,000	342,524	Napier.	Napier.
Grafton	7,350	12,000	343,780	Thames Co.	Humphreys.
Theseus	7,350	12,000	344,324	Thames Co.	Maudsley.
Bonaventure	4,360	9,000	237,262	Devonport.	Hawthorn Leslie.
Jason	810	3,500	49 068	Barrow Co.	Barrow Co.
Jaseur	810	3,500	49,138	Barrow Co.	Barrow Co.
Niger	810	3,500	48,918	Barrow Co.	Barrow Co.
Onyx	810	3,500	48,918	Laird.	Laird.
Renard	810	3,500	54,778	Laird.	Laird.
Circe	810	3,500	62,008	Sheerness.	Penn
Hebe	810	3,500	68,062	Sheerness.	Sheerness.
Leda	810	3,500	59,539	Sheerness.	Penn.
Alarm	810	3,500	59,346	Sheerness	Penn.

Jason.—On November 18th, 1892, Her Majesty's ship *Jason*, the first of six torpedo gunboats building by contract under the Naval Defense Act of 1888, had a preliminary trial in Morecombe Bay; this being satisfactory, the vessel was handed over to the navigating party appointed to receive her, and then proceeded to Sheerness, where her official trial has since taken place. The dimensions of all the six vessels are the same, viz.: Length between perpendiculars, 230 feet; breadth, extreme, 27 feet; depth, moulded, 14 feet 3 inches, with displacement of 810 tons at 8 feet 9½ inches load draught. They are flush-decked, with a forecastle forward extending one-third the vessel's length, officers' accommodation being aft on the lower deck The vessels have twin screws, with two sets of independent, triple-expansion, vertical, inverted-cylinder engines, capable of developing 3,500

I.H.P., and driving the vessel at a speed of 20 knots per hour, the diameters of cylinders being 22 inches, 34 inches and 51 inches, with 21 inches stroke. The engines are placed in separate engine-rooms, inclosed in water-tight bulkheads.

The boilers are of steel, four in number, 16 feet long by 7 feet 8 inches diameter. They are of the locomotive type, and each has two furnaces, the working pressure of steam being 150 pounds. The boilers are placed in two boiler rooms immediately before the engines, and are also inclosed by water-tight bulkheads. These gunboats having been designed for such a high speed, combined with large coal capacity, the machinery and boilers occupy nearly half the vessel's length. The engines and boilers are surrounded by bunkers with 3 feet of coal from below the load water-line to the upper deck, giving very efficient protection from shot and shell. The magazines are well protected, and the hull is divided transversely and longitudinally into numerous water-tight compartments.

In the eight-hours' natural-draft trial from Sheerness, the machinery worked most satisfactorily in all respects, the mean results being as follows: Vacuum, starboard, 24 inches; port, 24.5 inches. Revolutions, starboard, 216; port, 224. Indicated horse power, starboard, 1,305.4; port, 1,367.8. Collective indicated horse power, 2,673.2. The air pressure in stokehold was .8 inch, while the draught of the vessel was 8 feet 6 inches forward and 11 feet 6 inches aft; mean, 10 feet. The mean horse power up to the eighth hour was 2,723 indicated horse power, but as there was a good margin—the contract being for 2,500 indicated horse power—it was decided to allow the fires to burn down during the last hour, so as to give as long a time as possible for this process.

The official full-power trial took place at the mouth of the Thames on the 9th December, 1892. The engines and boilers were in charge of the contractors' representatives, and the trial was watched by officials representing the Admiralty, the Steam Reserve, and Sheerness Dockyard. The trial was attended with very successful results. The engines developed 3,540 horse-power, with an average speed of 20 knots. The trial was made under

forced draft, 2 inches of air pressure being used. The steam pressure in the boilers averaged 150 pounds, the engines keeping up a mean of 243 revolutions per minute.

RUSSIA.

Torpedo-cruisers Wojewoda and Possadnik.—The splendid results achieved by the torpedo-cruiser *Lieutenant Kasarsky* (see Vol. III, p. 281, of JOURNAL) built two years ago by Mr. Schichau, induced the Russian Government to order two new and similar cruisers, *Wojewoda* and *Possadnik*, and these have fulfilled all the conditions imposed by the contract. The principal dimensions and particulars of these ships are: Length, 190 feet 4 inches; breadth, 24 feet 3 inches; draught, aft, 10 feet 0 inch. The vessels are fitted with triple-expansion engines of 3,500 I.H.P. and two locomotive boilers. The armament consists of several quick firing and machine guns, one fixed torpedo-launching tube and two training tubes on deck. Electric light is fitted over the vessel, besides a projector. The speed per hour contracted for was 21 knots on a three-hours' continuous trial. The mean speed attained on the trial was 22.4 knots, the coal consumption on forced-draft trial per I.H.P. per hour being $1\frac{1}{2}$ pounds. The displacement on the trial with full coal bunkers (90 tons) and all weights and equipments was 400 tons. On a speed of 14 to 16 knots the coal on board will drive the ship a distance of 3,500 sea miles.

It may be remembered that the *Kasarsky* made the trip from Elbing round Europe to Nicolaief, and is now counted as one of the Black Sea fleet.

Rurik.—The new Russian cruiser *Rurik*, the largest warship yet constructed in Russia, and the largest of modern cruisers in the world, was launched on the 3d of November, 1892, from the Baltic Works on the Neva. The *Rurik* measures from stem to stern 435 feet, with 67 feet beam, 25 feet 9 inches depth and 10,933 tons displacement. Hitherto the largest ironclad in the Russian Navy has been the *George*. In the British fleet the largest are the *Blake* and the *Blenheim*, of 9,000 tons displacement. The longest existing armored warship after the *Rurik* is

the *Sardegna*, of the Italian fleet, which is 410 feet. The new cruiser embodies the culminating point of what may be called the cruiser policy, which the Russian Admiralty inaugurated, or at least strenuously worked at bringing to perfection. It is claimed for her that she will be able to make the voyage from Cronstadt to Vladivostock, on the Pacific, about 19,000 miles, without once coaling on the way. Her speed is calculated at 18 knots, with triple-expansion engines, made at the Baltic Works, of 13,250 indicated horse power. There are two sets of engines to each shaft. She will have a belt of compound armor 10 inches thick, and extending over two-thirds of her length. She will carry in her battery sixteen 6-inch guns, and on her upper deck four 8-inch guns, two forward and two aft, besides six guns of 4.8 inches and eleven machine guns.

ARGENTINA.

Libertad and *Independencia*.—Messrs. Laird Brothers, of Birkenhead, have in their yards two remarkable little ironclads being built for the Argentine Government. One, the *Libertad*, is practically complete, and will be presently commissioned by officers and men who have been brought for her from South America in the Argentine ironclad *Almirante Brown*. The other, the *Independencia*, is somewhat less advanced, but is of exactly the same type. The problem which was presented to the builders by the Argentine Naval Commission was the construction of a heavily-armed and well-armored ram, of fairly good speed, of light draught, and of the smallest possible displacement. The manner in which—thanks to the ingenuity of Messrs. Laird, and of their naval architect, Mr. Siemens—the various difficulties have been met is remarkable, for, in reply to the demand, there has been created a type of vessel which can, with all bunkers filled, carry 340 tons of coal. There is an 8-inch compound armor belt along about two-thirds of her length; substantial athwartship bulkheads, one 8 inches and the other 6 inches thick, and an overall protective deck; a partial double bottom, and about forty separate watertight compartments; two barbettes,

with armor varying from 5 inches to 8 inches thick, surmounted by steel shields of 5-inch plates.

There are twin screw, compound engines of 2,780 collective H.P., which have driven the vessel for four hours at a mean speed, in a heavy swell, of 14.21 knots. The radius of action is over 3,000 knots. The mean draught is 13 feet, and the displacement 2,300 tons.

The ship is also provided, in addition to two torpedo ejectors, for 18-inch Whiteheads, with two 9.37-inch Krupp guns, four Elswick 4.7-inch quick-firing guns, with shielded mounts, four 3-pounder Nordenfelt quick-firing guns, two 1-inch three-barrelled Nordenfelts, and two machine guns; with electric lights and search light, and with all the most recent improvements, including high-angle fire mounts for the two big guns, and excellent and well protected arrangements for hoisting powder and projectiles, and for loading by pneumatic machinery. The general design is that of a greatly reduced *Barfleur*, 230 feet long on the water line, and 44.4 feet broad, with a freeboard of 9.6 feet. There is a flush upper deck without bulwarks, a central breastwork, above which are stowed the boats, a 4-inch conning tower, and a single military mast with two tops. The heavy guns, which, though by Krupp, are on Elswick mountings, can be raised by the hand of a single man from their extreme depression to their maximum of 40 degrees in about 25 seconds, and everything connected with them can be worked by manual power. There are circulating pumps with a capacity of 600 tons per hour, auxiliary bilge and other pumps, separate combustion chambers to the boilers, and wide-spaced tubes, two powerful anchor capstans, balanced rudder, steam steering gear and duplicate electric plant; yet the ship is roomy, and the officers' quarters are even spacious. At her official trials, the *Libertad*, with natural draft, and an air pressure not exceeding ½ inch, made a mean speed of 13.35 knots for four hours, and with moderate forced draft, the air pressure being 1.05 inches, a mean speed of 14.218 knots for the same period. The maximum speed was 14.41 knots, and even this could, without any undue forcing or

danger to the tubes, have been bettered, but the contract speed had already been exceeded by nearly a knot in each case.

CHILE.

Capitan Prat.—This vessel, which has been recently completed by the Forges et Chantiers de la Méditerranée, La Seyne, for the Chilean Government, is 328 feet long, 60 feet 6 inches beam and 19 feet 9 inches deep, and is constructed of steel. Motive power of 12,000 I.H.P. with forced draft is supplied by triple-expansion engines. The armament of the vessel comprises four 24-cm. Canet guns of 36 calibers, mounted on Canet gun carriages and enclosed in barbette turrets; eight 12-cm. Canet quick-firing guns of 45 calibers; four 57-mm. Hotchkiss quick-firing guns; six 37-mm. Hotchkiss quick-firing guns and five Accles guns. Electricity has been adopted to a large extent as the motive power in the manœuvring of the guns and turrets. Three speed trials of the vessel have been carried out, the results of which are shown in the following table, the last trial being under forced draft:

Duration of trial.	Mean speed, knots.	No. of revolutions.	I.H.P.	Coal consumed per H P. per hour.
6 hours	17.42	104.18	8,493	1.34 pounds.
6 hours	17.825	110.591	9,648	1.575 pounds.
2½ hours	18.315	118.17	12,150	1.575 pounds.

JAPAN.

Yoshino.—There was launched on November 20th, from the Elswick shipyard of Messrs. Armstrong, Mitchell & Co., Newcastle, the powerful protected cruiser *Yoshino*, built for the Imperial Japanese Navy. The *Yoshino* has a length of 360 feet, a breadth of 46½ feet, a mean draught of 17 feet, and a displacement of 4,150 tons. She is constructed of steel throughout. Over the machinery, boilers, magazines and steering gear there is a strong steel protective deck, 4½ inches thick on its sloping sides, and 1¾ inches thick on its horizontal portions. The hull is subdivided throughout into numerous watertight compartments, most of these between the protective deck and the deck

immediately above it being used as coal bunkers. The total supply of coal which can be carried by the ship is just 1,000 tons, which will enable her to have a large and exceptional radius of action at cruising speeds. It is confidently expected that with the boilers working under forced draft the engine power of nearly 15,000 horses will be obtained. With this realized power, the speed of the ship will not fall short of 23 knots, and she will thus be the fastest cruiser afloat. The *Yoshino* will be armed solely with quick-firing guns of the latest and most approved pattern supplied by the Elswick firm. She will have four 6-inch quick-firing guns, and eight 4.7-inch quick-firing guns, and there will be, besides, an auxiliary armament of twenty-two 3-pounder quick-firing guns. Five torpedo tubes will also be carried. A conning tower of steel armor stands on the forecastle, and within it are placed all the neceesary appliances for working and conning the vessel in action.

TURKEY.

The Ottoman Navy.—The *Sultan* is paying great attention to his Navy, and now has under construction in Turkey an armored cruiser and six gun vessels. Eighteen more gun vessels are to be laid down, and two out of a number of torpedo boats have lately been developed by the builders, the Germania Co., of Kiel. The names of the new vessels are: The armored cruiser *Abdul Kader;* the six gun vessels, *Keshaf* (Explorer), *Saïki Shadi* (Joyous), *Seyah* (Traveler), *Nasri Hudra* (Divine Power), *Barikei Zafer* (Lightning of Victory,) and *Pelenki Deryah* (Sea Tiger); and the torpedo boats, *Sehaf* (Arrow), and *Perviz* (Triumphant).

MERCHANT STEAMERS.

Peru.—The steamship *Peru* was built by the Union Iron Works, of San Francisco, to carry passengers from San Francisco to Japan and China, and has accommodations for 104 first class, 75 second class and 600 steerage passengers.

The hull is built entirely of open-hearth steel under special survey to receive the highest class in Bureau Veritas, and in the United States Standard Steamship Owners, Builders and Underwriters' Association.

She has two masts schooner rigged, and has three decks and a hurricane deck. The main and upper decks are of steel. There are five water-tight bulkheads dividing the hull into six water-tight compartments, in one of which is placed the engines and in another the boilers.

The vessel is lighted by electricity, and is fitted with a dense-air refrigerating machine and a distilling plant for supplying fresh water for the boilers and for drinking purposes. The fresh-water tanks have a capacity of 15,000 gallons. She is also fitted with a steam windlass, steam capstan aft, steam steering engine, four steam winches and all modern conveniences for handling cargo.

She is propelled by a single screw, driven by a triple-expansion engine of 2,800 I.H.P., which is designed to give a speed of fourteen knots. The steam is supplied by six main and one auxiliary boiler, built of steel and of the cylindrical, horizontal fire-tube type, designed for a working pressure of 160 pounds per square inch. The coal bunkers have a capacity of 1,000 tons of bituminous coal.

Hull.—The principal dimensions of the hull are: Length over all, 346 feet; Length on water line, 326 feet; beam, 45 feet; depth of hold, 27 feet 6 inches; displacement, 5,010 tons.

Engines.—The engines are vertical, inverted cylinder, direct acting, with the high-pressure cylinder aft. Steam is admitted

to H.P. cylinder by a piston valve, and to I.P. and L.P. cylinders by slide valves, all worked by link motion.

The condenser is of cast iron, and forms part of the engine frame. The air pump, which is vertical and single acting, and two bilge pumps, are worked by a beam from L.P. crosshead.

The circulating water is supplied by a centrifugal pump driven by a compound engine with cylinders $7\frac{1}{2}$ inches and 12 inches by 6 inches stroke. The propeller is right handed and has four adjustable blades.

There are independent feed, fire and bilge pumps.

The principal dimensions are—

Diameter of H.P. cylinder	28 inches.
Diameter of I.P. cylinder	44 inches.
Diameter of L.P. cylinder	70 inches.
Stroke	48 inches.
Diameter of air pump	30 inches.
Stroke	24 inches.
Cooling surface	4,748.8 square feet.
Diameter of propeller	15 feet 6 inches.
Pitch, leading edge	21 feet.
Pitch, trailing edge	23 feet.
Pitch, mean	22 feet.
Helicoidal area	73.68 square feet.

Boilers and smokepipe.—The fire room is fore-and-aft, with three boilers on each side connected to one cylindrical smokepipe. Each boiler has two corrugated furnaces.

The principal dimensions of the main boilers are—

Diameter of boilers	12 feet 7 inches.
Length of boilers	12 feet 2 inches.
Diameter of furnaces	45 inches.
Heating surface in each boiler	1,591 square feet.
Grate surface in each boiler	49 square feet.
Thickness of shell	$1\frac{1}{8}$ inches.
Thickness of butt straps	$\frac{7}{8}$ inch.
Thickness of furnaces	$\frac{5}{8}$ inch.
Thickness of tube sheets	$\frac{3}{4}$ inch.
Diameter of smokepipe	8 feet 10 inches.
Height above grates	88 feet.

It is arranged that the ashes may be hoisted and dumped overboard, or may be expelled directly from fire rooms by an hydraulic ash ejector.

This consists of a 6-inch cast-iron pipe, leading overboard and discharging above the water line. Into the lower end of this pipe is the pump discharge through a nozzle $\frac{5}{8}$ inch diameter. A pressure of from 150 to 200 pounds per square inch is maintained in this pipe when the ejector is in use. Connecting with the discharge pipe, above this nozzle, is a hopper, with cover secured by clamps. To operate the ejector, the cover of the hopper is secured down and the pump started. As soon as the current is formed, the cover is lifted and the ashes shovelled into the hopper. The force of the discharging stream is such as to throw everything put into the hopper well clear of the ship's side.

The Trial Trip.—Under the contract, the *Peru* was to maintain a speed of fourteen knots for six hours. The engines had been turned but five hours before leaving the dock for the trial trip. For six hours after leaving the dock the engines were run at three-quarters speed, and at the end of that time the revolutions were increased to full speed. This speed was kept up for six hours. Four hours of the time was over the Government course, and the average speed for the four hours was 14.96 knots. Everything worked without the slightest hitch.

It was found that the draft was stronger than was needed, and at no time were all the ashpan doors down together. After the trial twelve feet of the smoke pipe was cut off. Four days after the trial was finished the *Peru* started on her regular trip and made the run to Yokohama at an average speed of fourteen knots per hour.

On the return trip this speed was slightly increased.

No trouble was experienced with the engines or boilers during the entire trip, and no water was used on the journals.—[Contributed by Passed Assistant Engineer Wm. B. Dunning, U. S. Navy.]

S. S. New York and Paris.—On February 25, the International Navigation Company's twin-screw steamship *New York*, formerly the well-known Inman liner, *City of New York*, will sail from the port of New York, and this event will mark the establishment of an American line of transatlantic steamers under the American

flag. Important changes will follow, and some of them are here enumerated: The Inman line ceases to exist, and the steamers which made it famous will have their names shortened by the removal of the appellation "City of," and simply be known as *New York*, *Paris*, *Berlin* and *Chester*. The house flag of the American line will consist of a blue American eagle on a white ground. Southampton will now be the English terminus of the line, and the arrangements made there will enable passengers to land at the new Empress dock directly from the steamship. A special train will take them direct to London in one hour and forty-five minutes. The *Paris* and the *New York* are the steamers under the American flag, but the company intends to build at Philadelphia five new steamships. Two of these, already ordered, will be larger than the *Paris* and the *New York*, and will be built by the Cramps. The three others will measure 10,000 tons, or only 500 tons less than the *Paris*. It is the intention of the new company to run weekly from New York one steamer direct for Southampton and one for Antwerp, calling at Southampton and Boulogne. The steamers will be the fastest first-class boats. The company will also run a line of steamships to Queenstown and Liverpool. Two of the fleet for the Liverpool route are under construction in England. They will measure 8,000 tons, and will carry freight and second-cabin passengers only.

International Line's Great Ships.—The general outlines of the plans for the new steamers being built by Cramp & Sons, of Philadelphia, for the International Navigation Co., have been determined upon, and the work of laying keel blocks and getting out material has commenced. As has already been stated in the "Sun," there are to be six ships in all. Four of the number are to be somewhat smaller than the *New York* and the *Paris*, and are to run on the line between New York and Antwerp. Two are to be larger than the *Paris* and the *New York*, and are to run with these vessels on the New York and Southampton line, making four steamers on each line.

The engines of these new vessels will be their most interesting features, on account of the great power they are to develop and

the high speed they are to give the ships. All the vessels will have twin screws, with the engines in separate water-tight compartments. The smaller ships will have about 17,000 horse power and are expected to have about the same speed as the *Paris* and the *New York*. The larger ones are to have about 25,000 horse power and are expected to be faster than any vessels afloat, including the two new Cunarders now being built on the Clyde. All the engines are to be of the vertical, inverted, direct-acting quadruple-expansion type, with a working pressure of 210 pounds of steam to the square inch at the boilers.

Their performance will be watched with much interest, for the reason that this will be the first time there has been any effort made to use quadruple expansion in engines of over 4,000 horse power, and in only one or two instances has it been applied to engines of that power. The advantage of quadruple-expansion engines is in the saving of fuel, which will amount in these new ships to about ten per cent. over triple expansion, and for the larger ships will be about fifty tons per day. Their coal consumption is estimated at about 400 tons per day each. The difficulty in applying quadruple expansion to engines of large power is due to the difficulty of making boilers strong enough to stand the tremendous pressure required. In small vessels, where water-tube boilers can be used, this difficulty can be easily overcome, but there is doubt about the water-tube boilers working well on a large scale, and in the new Inman steamers the ordinary type of Scotch boilers will be used, but they will be built of heavier material than has ever been used in boilers before, and will be somewhat less in diameter than many of the large boilers now being built for triple-expansion engines of high power.

In this connection the trial trip and the subsequent cruising of the coast-defense vessel *Monterey* will be watched with great interest, for Commodore Melville is making in her the first experiment with water-tube boilers in a large vessel. He intends to give her a very severe trial so as to ascertain the efficiency and durability of this type of boilers. If the expectations of her builders are realized, one of the results will probably be a very

general adoption of water-tube boilers and quadruple-expansion engines for large vessels.—"Baltimore Sun."

Express Passenger Steamers for the Lakes.—The Northern Steamship Company, in preparation for the World's Fair and the summer of 1893, is adding to its fleet exclusive passenger facilities that will startle the Great Lakes. The company is now having built at the Globe Iron Works, of Cleveland, two of the largest, swiftest and most elegantly appointed steamships ever built to run on American waters. No expense will be spared to make these splendid passenger steamships as safe as human foresight can secure and as comfortable as human ingenuity can devise. They are but the commencement of a fleet of six steamships ultimately to be built, and they are to be run on the Buffalo and Duluth route in connection with the great Northern Railway line as part of that great system. The first two vessels already referred to, it is hoped to complete in time to be running on their regular route, during the latter part of the season of 1893, leaving each terminal twice each week. The speed of these greyhounds will average about twenty miles an hour, and the distance between Buffalo and Duluth, say 1,110 miles, will be traversed in about sixty hours, including several hours' stop at Cleveland, Detroit and Sault Ste. Marie. The rational system of separating the transportation charges from the charges for sleeping berths and subsistence will be adopted by this company. A superbly-appointed restaurant located on the main deck forward will be a special feature.

These vessels are of a type entirely new to the lakes, as they will cost no less than $600,000 each, and are patterned in every way after the best design of ocean-going passenger boats. They are 360 feet on the keel, 380 feet over all, 44 feet beam and 34 feet deep. In the many new features as to boilers, engines, cabin arrangements and provision for lightening and increasing draught, the construction of these boats will, however, cause a great deal of discussion, as they will represent the latest practice known in shipbuilding.

Each of these steamers will have two overhead-cylinder, vertical, quadruple-expansion engines, which will develop at least 7,000

horse power. The high-pressure cylinder is forward, followed by first and second intermediate and the low pressure. The sizes of these cylinders are 25, 36, 51½ and 74 inches, the stroke being 42 inches. The four crank shafts will consist of four interchangeable parts. The cylinders will face outboard, and will be supported by wrought columns in front, and forked cast columns behind, with slipper guides. There will be piston valves throughout and Joy valve gear. There will be one piston valve for high-pressure cylinder and all the others will have two each. The air pumps and condenser will be independent. The air pump will be vertical, and have two single-acting cylinders 36 inches diameter by 16 inches stroke. The twin screws will be 13 feet diameter by 18 feet mean pitch. The steam will be furnished by 28 Belleville boilers at 195 pounds pressure.

An idea of the massive proportions of the engines will be suggested by the weight of the bed plates, which will weigh 34 tons each. The low-pressure cylinders will each weigh 24,000 pounds. The pistons of the low pressure and second intermediate cylinders will be of cast steel.

There is, of course, no way of determining just what the fuel consumption of a big steamer of this kind would be on the very fast run of about 1,000 milies from Buffalo to Duluth, but such questions are always matters of gossip, and it is estimated that the supply required for the trip will be 800 to 1,000 tons. In the engine room department it is probable that no less than sixty men will be required. These items of expense together with the very heavy first cost of the steamers, and the fact that they will have little more than three months of profitable service during the summer season, with no revenue from freight, have caused considerable comment as to the success of this undertaking. This comment is, however simply an expression of opinion which is certainly not based upon a thorough canvass of the subject. On the other hand a line of this kind has been talked of for years among railway managers and vessel owners well posted on the passenger business, and it is more than probable that the owners of the steamers are thoroughly satisfied as to patronage. It is even hinted that their present business with connecting lines is sufficient assurance of success in this enterprise.

American capital and American material and workmanship will control every part of the building of these ships. All of the machinery and even the boilers, which are of French pattern, will be constructed by the Globe Iron Works Company. Of the material in the hulls, the angles will come from the Cleveland Rolling Mill Company, plate from the Otis Steel Company, and beams from the Pencoyd Iron Works.—[Compiled from "Seaboard" and the "Marine Review."]

Christopher Columbus.—This ship is built to carry passengers from Chicago to the World's Fair grounds, a distance of between six and seven miles. She will have accommodations for from 4,000 to 5,000 passengers at one time, and will thus be the largest excursion steamer afloat. Before the critical eyes of the more progressive people of the entire world, who will be sure to visit the fair, the whaleback steamship, *Christopher Columbus*, is sure to create more discussion and to be the biggest sensation in the maritime world of any event since the construction of the ark.

She is built entirely of steel, is 362 feet long over all, 42 feet beam and 24 feet deep. She will be supplied with a triple-expansion engine built by Samuel F. Hodge & Co., of Detroit. The cylinders are 26, 42 and 70 inches in diameter, with a piston stroke of 42 inches, to develop an indicated horse power of 2,600. Steam, at a maximum pressure of 160 pounds to the square inch, will be supplied from six cylindrical boilers, built by the Cleveland Shipbuilding Company, 11 feet in diameter by 12 feet long, each containing two corrugated furnaces 44 inches in diameter. She will be propelled at a maximum speed of twenty statute miles an hour by a single screw 14 feet in diameter, with a pitch of 19 feet. Although built upon the ordinary whaleback principle, a departure is made in the location of her engines and boilers, which will be amidships. There will be seven turrets upon the main deck of this steamer, all of which will be $7\frac{1}{2}$ feet in height and elliptical in shape, the forward one, 19 by 13 feet in size, containing the windlass, with a door at the after end, and lighted with six large bull's eye dead lights; turrets Nos. 2 and 3 are 26 by 18 feet, with two doors at after end, which lead to

stairways to saloon deck above and between decks below, these turrets containing 14 dead lights; No. 4 is 27 by 18 feet, and contains the smokestack, air fans, ash hoists, etc., is supplied with doors on each side of after end, and is lighted with 10 dead lights. This turret contains entrance to fire room, firemen's wash room, etc. The next turret aft (No. 5) is 27 by 18 feet and encloses the engine room and machinery, having a door upon each side and lighted with 14 dead lights. The two after turrets are respectively 26 by 16 and 22 by 12 feet in size, the former containing two doors at after end and 14 dead lights, and the aftermost turret having two doors at forward end and 10 dead lights. These turrets also lead to stairways similar to the two described forward. In addition to these she has four athwartship gangways on each side, three of which are 7 feet in the clear and one 5 feet in the clear, all being 6 feet high, through which passengers enter and leave the ship. She also is provided with side hatches for cargo purposes. Extending fore-and-aft are two tiers of glass bull's eyes and square dead lights, located, of course, to admit light into the between decks. Amidships are dining and refreshment tables and booths, and forward are situated the kitchen, mess room and crew space.

She has a double bottom, 42 inches deep, extending fore-and aft, with tanks in the fore and after peaks, affording a total water ballast capacity of 730 tons. These water bottoms are built on the McIntyre principle, with 9 girders about 3 feet apart, exclusive of the tank margin, and are braced with angle irons and brackets to the floor plates, which braces are two feet apart. She is built on the web frame principle, the webs being spaced 16 feet apart, except at boiler space where they are 12 feet apart, and the engine room where they are 8 feet apart, in both of which latter plates the girders are double bracketed. She is subdivided with 9 water-tight bulkheads, all athwartships, in addition to which, extending from her nose in the fore peak aft, 42 feet, she has a fore-and-aft bulkhead. Her lower deck beams are of channel iron stayed by I stanchions. In the between decks the stanchions are $4\frac{1}{2}$-inch tubes riveted to a girder underneath the shell deck of double angles. In addition to this there is

another girder on each side of the same section, extending fore and aft. Her between decks are of steel, flush riveted, extending all fore and aft, with T iron fore-and-aft butt straps, the standing-down web forming a splendid stiffener for the fore and-aft-seams. In the hold are two stringers, extending all fore and aft, braced with face angles, besides another stringer with double angles back to back. Outside are two steel fenders riveted to the hull, which are 12 inches in diameter. She has a flat plate keel of 29¼-pounds and the shell for one half her length amidships is 24 pounds, tapered to 18¾-pounds at ends, with bilge strake and sheer strake of 26½-pounds for half the length tapering to 18¾-pounds. The shell above the sheer strake is 21½-pounds, with shell deck of 18¾-pounds. The shell deck is flush riveted with seams and butts, forming an entirely smooth surface. The shell butt straps are 3 pounds heavier than plates and are treble-riveted for one-half her length amidships.

The steel saloon deck, 11 pounds in thickness, with a 16-pound strake at each side of turrets, the whole being efficiently connected to heavy angle iron rings on each turret, is connected to the ventilator tubes with angle iron collars, and is also fitted with angles at each side of the tubes running fore and aft, thereby producing a bracing quite unusual, and particularly effective. These ventilator tubes are 9 inches in diameter and located 12 feet apart almost the entire length of the ship. The saloon deck is sheathed with wood, and the saloon proper is 225 feet long by 30 feet wide. It is provided with three double doors on each side, which, when open, leave a span of 4½ feet in the clear.

The deck outside the saloon will consist on either side of a space four feet in width, at either end there being an entirely unobstructed space of 32 feet in length. The ventilator tubes form the side supports to this saloon deck, and come through the joiner work, the framing of which is bolted clear through the ventilators fore and aft. This will not only stiffen the sides of the saloon, but, being prettily decorated, will form handsome columns both inside and outside the saloon. All stairways to the saloon deck are five feet wide in the clear, and are ample for the quick loading and landing of passengers.

Over the saloon is the promenade deck, 257 feet long, in the center of which is a skylight 15 feet in width, and extending a distance of 138 feet, with a large glass dome over the after stairway. The smokestack and fire-room ventilators go through the skylight. At the forward end of the skylight is situated the "texas," which contains the grand staircase from the saloon, captain's cabin, pilot house, wheelmen and mates' quarters. Besides this there is a dome skylight over the forward staircase.

The lifeboats are sixteen in number, each 20 feet long, and are arranged on each side of the ship, each alternate ventilator tube being provided with a swinging crane which runs up and forms the air downcast, and also are intended to swing the boats as required.

The hold contains a cross bunker, the full breadth of the ship and extending a distance of 24 feet fore and aft.

She will be supplied with all necessary life-saving equipment and is expected to accommodate 5,000 passengers on the route between the great pier in front of the Auditorium in Chicago and the large pier built at Jackson Park at the World's Fair.

Besides this, she will be used for making night excursions up the lake.

The whaleback type of structure has been in use now for a period of four years, but altogether as a freight vessel. In that time those built have been seen upon both oceans, upon the lakes, have grown to a vast fleet, and are beyond all question a pronounced commercial and financial success. They have been so often and so fully described as to scarcely need extended reference to their departure from the ordinary craft, from which they differ only above the water line, with the exception alone of their spoon-shaped blunt-ended conoidal bows. They have been so fully illustrated, as well, as to show their peculiarities above the water line. Briefly, however, and for the information of those of our readers who are unfamiliar with the whaleback vessel, they may be described as oval steel structures with flat bottoms, and with girders running fore and aft, to which the frames are attached, which frames are also oval shaped and extend in one piece from the keelson clear around the shell of the ship, ending only at the opposite side of the keelson.

It is a well known and thoroughly proven mechanical fact that round and oval structures will bear more strain and inherently possess greater strength than any other form, and this fact is utilized to perfection in the whalebacks. They are all provided with several athwartship bulkheads and a water bottom, as already stated in the foregoing description of the *Christopher Columbus*. The shell plates follow the frames, and as a consequence their decks present a rounded surface, sloping inboard from the sheer strake to the center. Upon this deck are placed removable steel stanchions, to which are attached life lines, forming a protection similar to the bulwarks of the old-fashioned ship, and which in these whalebacks has been found sufficient upon all seas for the safety of the crews. In the ordinary vessels of this type there are usually two large round turrets in the after end of the ship which pierce the deck, and which accommodate the engines and boilers, smokestacks, etc. Upon the top of these a deck is fastened, and upon this deck are placed the dining room, galley, officers' rooms and the pilot house at the forward end, and the deck above that is used simply to stow lifeboats, rafts, etc. Forward is another turret, in which is placed the windlass, forward of which, inside the hull, are the quarters for the crew.

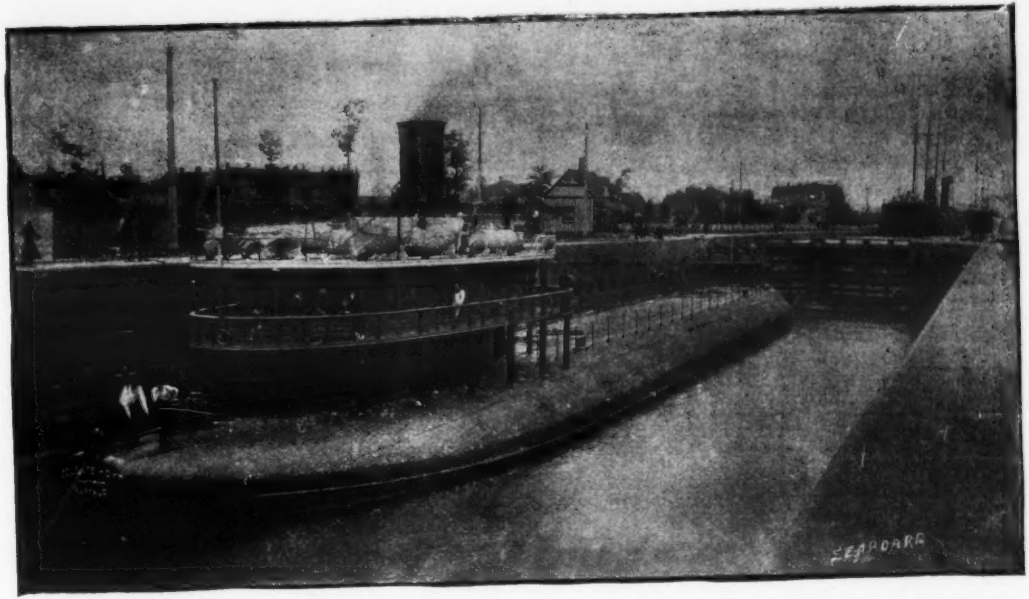
In closing this reference to the ordinary whaleback freighter, it is proper here to say that the features just described have produced the following results in use:

1. Their construction above the water line is such that the sea never breaks over them, simply rolling up to and receding from the sloping sides, with an occasional wash in particularly heavy seas.

2. They admit of immersion to the point of being almost awash, and ten per cent. of buoyancy makes them better sea boats than when possessed of greater freeboard.

3. They have been found to ride the sea in storms that other vessels are unable to make headway in, and this is owing to the fact that the seas do not break over them, and that but comparatively little superstructure opposes the wind.

4. Everything above the water line being rounded—decks,


SEABOARD

turrets, stanchions, etc.—should the seas ever strike any part, they would be deflected and dissipated without the possibility of injuring anything.

5. These conditions have enabled them to carry more cargo per ton of displacement than other vessels, and with less expenditure of power and less consumption of fuel.

6. The speed of the whaleback steamer, as compared with other vessels equal in size, power, capacity and draught, is greater.

7. Their construction is much cheaper than ordinary vessels, for the reasons enumerated, and the cost of maintenace is much less than upon other craft, because there are no spars, rigging, bulwarks, etc., to keep up and renew.

8. They are manned with smaller crews than are necessary upon other vessels of the same size.

It is thus apparent why these vessels are successful commercially and financially, nothwithstanding the fact that as compared with the ordinary lake carrier, they are built of heavier material all through, including frames, girders, plates, beams, etc., and have passed the severest tests of the American Shipmasters' Association, and are classed, upon the Atlantic coast, as well as upon the lakes, with the best ratings obtainable for any vessel.—[Reprinted and abridged from "Seaboard."]

[In this connection, the Council are glad to be able, through the courtesy of "Seaboard" and the "Marine Review," to publish the accompanying illustrations of whaleback freight steamers, which show their appearance before launching, and also when in regular service and fully loaded.]

Pioneer—Report of Engineer's Trial.—The trial was made for the purpose of ascertaining the maximum power that could be easily maintained for continuous trips, and the results of the Howden forced-draft system, with which this ship was fitted. No attempt was made to force the machinery beyond this point. The trial was made September 19, 1892, on Lake Erie, from Bar Point Light to Buffalo Breakwater, a distance of 230 miles. Indicator cards were taken every two hours, also readings from the gauges, showing air pressure in ashpit, boiler pressure, vacuum

and revolutions. The fan was kept as nearly as possible at a uniform speed of 400 revolutions per minute. The temperature of chimney was taken every four hours by a pyrometer, at a point of about nineteen feet above the tubes, which was the only accessible place. The only means of ascertaining the amount of coal used was to measure the amount left in bunkers at the end of the trial, and deduct this from the amount on board at the commencement of the trial, so that results given are only approximate.

VESSEL.

Length over all	241 feet.
Length keel	225 feet.
Breadth	35 feet 2 inches.
Depth	17 feet 6 inches.
Mean draught at trial	14 feet 8 inches.
Displacement at trial, net tons (2,000 pounds)	2,685.
Cargo and fuel, net tons	1,810.

ENGINE.

Cylinders, diameter	20 inches, 33 inches and 54 inches.
Cylinders, stroke	42 inches.
Propeller diameter	12 feet 6 inches.
Propeller pitch	14 feet 9 inches.

BOILERS.—FITTED WITH HOWDEN'S SYSTEM OF FORCED DRAFT.

Number	2.
Diameter	12 feet.
Length	11 feet 6 inches.
Number of furnaces, total	4
Diameter of furnaces	3 feet 7 inches.
Grate surface, square feet, total	164.
Heating surface, square feet, total	6,204.
Ratio	37.7
Diameter of fan	48 inches.
Area of air main, square inches	576.
Size of fan engine	7 inches by 7 inches.

PERFORMANCE.

Boiler pressure, pounds	159.27
Vacuum, inches	21.86
Revolutions per minute	87.90
Indicated horse power, H.P. cylinder	300.
Indicated horse power, Int. cylinder	386.
Indicated horse power, L.P. cylinder	421.
Indicated horse power, total	1,107.

Distance run during trial, miles	230.
Running time	17 hours and 40 minutes.
Speed of vessel, miles per hour	13.02
Slip of propeller, per cent	11 6
Coal per hour, pounds	1,867.8
Coal per hour, per I.H.P., pounds	1 68
Coal per hour per square foot of grate	22 68
Heating surface per I.H.P., square feet	2.79
I.H P. per sqare foot of grate	13 46
Air pressure in ashpit, inches of water	.563
Revolutions of fan per minute	400.
Revolutions of fan engine per minute	182.8
Temperature of chimney, Fahr	575°.
Temperature of feed water, Fahr	118°.

MAXIMUM PERFORMANCE.

Boiler pressure, pounds	165.
Vacuum, inches	21.5
Revolutions per minnte	90.
Maximum power developed, H.P. cylinder	317.
Maximum power developed, I.P. cylinder	433.
Maximum power developed, L.P. cylinder	473.
Maximum power developed, total	1.223.
I.H.P. per square foot of grate surface	14 86

The electric light engine was running for six hours, and steering engine constantly during trial. The last may seem unimportant, but owing to slack in trick gear and tiller chains, the engine had to make a considerable number of revolutions.

The *Pioneer* was built by the Detroit Dry Dock Company, ship and engine builders, Detroit, Michigan.

Campania.—The Fairfield Shipbuilding Co. are making steady progress with the completion of the *Campania*, the first of the two big Cunarders. The funnels of these vessels are from the top of the smoke boxes 120 feet in height, and their outside diameter is 20 feet. The electric installation on board is in keeping with the other details of these large vessels. There are four sets of generating plant in each ship, and each consists of a Siemens dynamo coupled direct to a Belliss engine, which runs at the rate of 280 revolutions a minute, and gives an output of 42,000 watts. This is capable of supplying the 1,350 16-c. p. incandescent lights, including eight large reflectors of eight lights each for working cargo throughout the ship, and in addition, a

powerful search light for facilitating the navigation of the ship into port, the picking up of moorings, and scouting in time of war. The large switchboard for controlling the light consists of 13 sections so arranged, that each may be connected with any of the four dynamos. From these dynamos and this large switchboard there runs through each ship an enormous length of wire, upwards of 40 miles, as a matter of fact. The supervision of the works is in the hands of Mr. J. T. Bottom', F. R. S.

Stern-Wheel Steamer for the Argentine Government.—A stern-wheel steamer has been built for the Argentine Government to the designs of Mr. Josiah McGregor by Messrs. Bow, McLachlan & Co., Paisley. The hull and machinery are arranged on McGregor's system, which differs entirely from that of the usual stern-wheel steamer. The engines and boilers are placed together near the middle of the vessel, and the power is transmitted to the wheels by means of long piston rods, led along the deck and enclosed in a box. In the case of larger vessels the cylinders are placed at the bottom of the vessel, acting diagonally, and the piston rods are boxed in under the deck. The arrangement completely eliminates some of the objectionable features in the ordinary type of vessel. The long steam pipe is avoided entirely, and from the central position of the bunkers the great variation of trim, consequent on the consumption of three or four days' fuel supply, is overcome, while the machinery, being placed where the vessel can best sustain its weight, admits of a better shaped hull, and the reduction of such structural appendages as kingposts, &c. It also secures the constant control and management of the engines and boilers by one man. The vessel is propelled by a divided feathering wheel, and, as the rudder is placed abaft of the wheel, the steering is a great improvement on that of the ordinary type. In this case a rudder was placed both before and abaft the wheel to secure equally good steering when the vessel was going ahead or astern, as she is to navigate rapids. In other cases, however, this forward rudder has been omitted.

The high-pressure cylinder is 18 inches in diameter, and the low-pressure cylinder 31 inches in diameter, each with a 3-foot

6-inch stroke. They are placed one in advance of the other, to get the piston rods close together. These rods pass through guide blocks placed at 1 foot interval, and inclosed in the box casing on deck. The objections at first raised to this have proved without foundation in practice, since the rods do not give any trouble. The engines are fitted with a somewhat novel application of Joy's valve gear, as it derives its motion from the piston rods and not from the connecting rods in the usual way. The air pump is worked by a bell crank from the main engine, and the circulating water is supplied by a 15-inch centrifugal pump and engine. The outside casing of the condenser is made of steel plate. There are 593 square feet of cooling surface. Steam is supplied by two boilers of the locomotive marine type, working under forced draft. They are somewhat large for the work required of them, on account of the indifferent fuel obtainable. They have a heating surface of 1,002 square feet and a grate area of 30 square feet, and are constructed to work with a pressure of 150 pounds per square inch. Air is supplied to the closed stokehold by a 4-foot fan and engine.

The hull, which measures 125 feet over all by 22 feet by 4 feet 6 inches side, is constructed of steel, 5 pounds per foot throughout, and is divided by five transverse bulkheads, and a longitudinal bulkhead which is water-tight except in the engine room. The upper deck framework is utilized to give strength to the structure, and is secured by pipe stanchions and diagonal rods to the hull. It terminates at, and is firmly secured to, the sides of the paddle boxes.

The light draught of this vessel, complete, with steam up and a full equipment of stores, is 20½ inches. At the trial it was 23¾ inches, at which displacement was 101 tons. The engines indicated 200 horse power, the steam pressure being 145 pounds, vacuum 24 inches, and number of revolutions 40½. The speed was 12¼ miles per hour.

Fast Small Yacht.—Mr. C. D. Mosher, the designer of the *Norwood*, is now completing a 78-foot boat, with 9 feet 6 inches beam. The designs are all Mr. Mosher's personal work, and certain novel details of the boiler and engine are covered by his

patents. The engine is of the quadruple-expansion type, with overhead cylinders supported on a base of cast and wrought iron by means of slender steel vertical pillars, each pair of which are braced with straining rods in the form of an X split down through the point of crossing, and provided with a screw, by which the braces can be strained until all racking is obviated. The stroke of the engine is ten inches, and the cylinders are respectively $9\frac{1}{2}$, $13\frac{1}{2}$, 18 and 24 inches in diameter. Every ounce of superfluous metal has been removed from the castings forming the cylinders, and they are handsomely jacketed with sheet brass. It is estimated that the complete engine will weigh less than 3,600 pounds, and that at a speed of 500 to 600 revolutions it will develop from 500 to 600 horse power, with a steam pressure of 250 pounds. To secure the maximum of weight with the maximum of strength, all of the working parts have been reduced to the smallest practical dimensions, or else relieved of superfluous metal at the center by boring. The rock shafts have one-inch holes through them. The piston and connecting rods are hollow, and the crank shaft has been bored out whenever a tool could be used upon it. This shaft was made from solid-steel forging weighing 2,012 pounds. The engine when set up will occupy less than 14 square feet of floor space.

The boiler is a pipe boiler, of peculiar construction, and is built with a view of standing great pressure, occupying little space and steaming rapidly.

BOOK REVIEWS.

LOGARITHMIC TABLES.—By PROF. GEO. W. JONES, of Cornell University.

In these days, when every engineer has become accustomed to the use of improved slide rules, pocket calculators, like Boucher's, and other mechanical aids to computation, ordinary methods, by the use of tables of logarithms, seem slow and laborious, and there seems danger that accuracy may be sacrificed to rapidity and reduction of labor.

In any case the computer wants his tables so arranged as to give him the least labor and the greatest insurance against error. Too often tables, valuable in themselves, are little used because they are poorly arranged and inconvenient.

The tables by Prof. Jones are excellent examples of careful arrangement, with every point looked after to make them convenient and accurate. They are distinguished by excellent type so arranged as to catch the eye at a glance and prevent confusion. By arranging the logarithms in blocks of five, separated by spaces instead of lines, the proper one can be located, after a little practice, by its position, almost without reference to the "arguments." This plan is followed in all the tables throughout the book.

The contents include four and six place logarithms of numbers and trigonometric functions, natural or Napierian logarithms, squares and cubes, and square and cube roots and reciprocals.

An excellent feature is a collection of such constants as specific gravities, specific heats, comparison of thermometer scales, melting points, and coefficients of expansion, besides which there are numerous "conversion tables" from metric to English units and *vice versa*.

The tables have been specially prepared for scientific use in

laboratories, and embody the result of Prof. Jones' long experience as a teacher of mathematics at Cornell University.

Altogether, the book is an excellent one, neatly bound and of convenient size. It is published by Prof. Jones, at Ithaca, N. Y. Price, $1.00 by mail, prepaid, or 75 cents by express at purchaser's cost.

ELECTRICITY AND MAGNETISM.—BY PROF. EDWARD J. HOUSTON, A. M.

ORIGINAL PAPERS ON DYNAMO MACHINERY AND ALLIED SUBJECTS.—BY JOHN HOPKINSON, M. A., Dr. Sc., F. R. S.

These two works, in small octavo, are published by the W. J. Johnston Company, 41 Park Row, New York City, at $1.00 each.

The first is one of a series of "Advanced Primers of Electricity," and is a simple and elementary exposition, in a comprehensive way, of the principles of electricity. It does not aim to be exhaustive, nor to cover the advanced applications of electricity. In the field it covers it is excellent.

Dr. Hopkinson's book, as its name implies, is a collection of original papers previously published in fragmentary shape. His eminence in electrical science assures the value of anything he writes. The papers here collected treat both of the theory and practical testing of dynamo machines, and contain many valuable points.

ANNUAL MEETING.

The *Annual Meeting* of the Society was held in the office of the Engineer-in-Chief of the Navy on December 26, 1892, and was a success in every way.

The election of officers resulted as follows:

President: Chief Engineer H. Webster, U. S. N.

Secretary-Treasurer: Passed Assistant Engineer W. M. McFarland, U. S. N.

Council, in addition to President and Secretary:

Passed Assistant Engineer F. H. Bailey, U. S. N.; Passed Assistant Engineer I. N. Hollis, U. S. N., and Assistant Engineer W. W. White, U. S. N.

The report of the Secretary-Treasurer for the year ending with the meeting was submitted, and shows the Society to be in excellent financial condition. Since the meeting it has been audited and approved. It is as follows:

WASHINGTON, D. C., *December* 22, 1892.

To the American Society of Naval Engineers.

Gentlemen: In accordance with the requirements of section 14 of the By-Laws, I have the honor to submit the following report of the finances of the Society to date:

Received from Assistant Engineer E. Theiss, late Treasurer	$865 25
" for annual dues	1,931 38
" for sale of extra copies of JOURNAL	78 95
" for sale of special reprints	61 55
" for advertising	1,162 50
" for interest on deposits, &c	14 53
Total receipts	$4.114 16
Expended for printing JOURNAL	$1,771 23
" for illustrations	369 55
" for stationery, postage and incidentals	187 12
" for Secretary's salary and office furniture	638 00
Total expenditures	$2.965 90
Balance on hand December 22, 1892	$1,148 26

All of which is respectfully submitted.

WALTER M. MCFARLAND,
Passed Assistant Engineer, U. S. Navy,
Secretary-Treasurer.

The paper by Passed Assistant Engineer Ira N. Hollis, U. S. N., which is published in this number, was then read; after which it was discussed by Engineer-in-Chief Melville, Chief Engineer Baird, Chief Engineer David Smith, and others.

It was decided to hold monthly meetings of the members in and near Washington until the advent of warm weather, these meetings being for the reading and discussion of professional papers. It was announced that the first of these papers, to be read at the meeting in January, would be one on "The Design of Air Pumps," by Passed Assistant Engineer Frank H. Bailey, U. S. Navy.

After a vote of thanks to the officers of the past year, and to Passed Assistant Engineer Hollis for his interesting paper, the meeting was entertained by remarks bearing on the progress in marine engineering by Clark Fisher, Esq., late Chief Engineer, U. S. Navy, and Irving M. Scott, Esq., Vice President and General Manager of the Union Iron Works. Mr. Scott commended most heartily the work of the Society, and urged that every effort be made to bring out new ideas and the fullest discussion of proposed improvements, in order that the present splendid position held by our naval vessels should be maintained.

OBITUARY.

ARTHUR T. WOODS.

Another young engineer whose professional career had only fairly begun has passed away in the person of Prof. Arthur T. Woods, who died February 7th at Chicago, of typhoid fever.

Prof. Woods was born at Minneapolis, Minnesota, January 9, 1859, and entered the Naval Academy as a Cadet Engineer in the fall of 1876. After maintaining a high rank in his class, he graduated in June, 1880.

He served on the U. S. Steamers *Galena* and *Quinnebaug* on the European Station, and on his return was promoted and commissioned as an Assistant Engineer in 1882. After a short tour of duty at the Navy Department in the Bureau of Steam Engineering, he was detailed as Professor of Mechanical Engineering at the University of Illinois, where he remained three years. He was then ordered to the U. S. S. *Trenton*, but the University authorities made such a flattering offer that he resigned and became the permanent professor of mechanical engineering.

In 1891 he accepted a call to fill the chair of Dynamical Engineering at Washington University, St. Louis.

For some time he had paid special attention to the problems connected with the economics of locomotive engines, and last year he resigned his chair at Washington University, moving to Chicago, where he became one of the editors of the Railroad Gazette, and also opened an office as a consulting mechanical engineer.

Prof. Woods had written a good deal on mechanical subjects, both for the technical papers and for text books.

He leaves a widow, the daughter of the late Admiral deKrafft, U. S. Navy, but no children.

BY-LAWS.

1. The Association shall be known as the AMERICAN SOCIETY OF NAVAL ENGINEERS.

2. The object of the Society shall be to promote a knowledge of naval engineering by the reading, discussion and publication of papers on professional subjects; by the bringing together of the results of experience acquired by engineers in all parts of the world, which, though valueless when unconnected, tend much to the advancement of engineering when published together in a JOURNAL of the Society, and by the publication of the results of such experimental and other inquiries as may be deemed essential to the advancement of the science.

3. The officers of the Society shall be a President, a Secretary and Treasurer, and a Council, all of whom shall be elected annually.

4. The Society shall be composed of Members, Associates and Honorary Members.

5. Officers of the Engineer and Construction Corps of the Navy, and persons in civil life who were formerly officers in said Corps, shall be eligible as Members.

6. Persons in civil life whose knowledge of engineering is such that they can co-operate with naval engineers in the promotion of professional knowledge, or who are intimately connected with the engineering profession, shall be eligible as Associates.

7. The Secretary of the Navy, the Assistant Secretary of the Navy, the Chief of the Bureau of Steam Engineering of the Navy Department, all ex-Chiefs of the Bureau of Steam Engineering, and such other persons as the Society may elect, shall be Honorary Members.

8. Members shall be admitted upon application upon payment of the annual subscription.

9. An Associate may be admitted upon application in writing, and upon payment of the annual subscription, provided his application have the recommendation of one Member, and receive the approval of a majority of the Council.

10. Associates shall be entitled to all the privileges of Members, except voting and holding office.

11. To be eligible as Honorary Member the candidate must receive the unanimous vote of the Council before his name shall be presented to the Society; and the favorable vote of two-thirds of the Members voting shall be necessary for election.

12. The direction and management of the affairs of the Society, and the editing and publishing of the JOURNAL, shall be vested in a Council composed of five Members, two of whom shall be the President and the Secretary and Treasurer. The Council may appoint one of its number Librarian.

13. The President shall preside at the annual meeting, and at the meetings of the Council.

14. The Secretary and Treasurer shall conduct the correspondence of the Society, receive subscriptions and pay all bills certified by the President. He shall submit an annual statement of the receipts and expenditures of the Society, which shall be audited by three other members of the Council.

15. The Council shall have authority to fill vacancies which may occur during the year.

16. Subscriptions, and all matter intended for publication, shall be sent to the Secretary and Treasurer; but no paper shall be read before the Society at the annual meeting, nor anything published in the JOURNAL. without the approval of the Council.

17. In deciding matters pertaining to the Society, none but Members shall be entitled to a vote.

18. The Society shall meet annually in the city of Washington, and at such other places as meetings can be conveniently arranged.

19. The annual subscription shall be five dollars, payable in advance.

20. Copies of the JOURNAL may be sold by the Society at a price which shall be fixed by the Council.

21. Amendments to these rules, or new rules, may be made at any time, provided all the Members are notified of the proposed change, and that a majority of those voting are in favor of the change.

AMERICAN SOCIETY OF NAVAL ENGINEERS.

OFFICERS FOR 1893.

President:
Chief Engineer **H. Webster**, U. S. Navy.

Secretary and Treasurer:
Passed Assistant Engineer **Walter M. McFarland**, U. S. Navy.

Librarian:
Assistant Engineer **W. W. White**, U. S. Navy.

Council:
Chief Engineer **H. Webster**, U. S. Navy.
Passed Assistant Engineer **Walter M. McFarland**, U. S. Navy.
Passed Assistant Engineer **Frank H. Bailey**, U. S. Navy.
Passed Assistant Engineer **Ira N. Hollis**, U. S. Navy.
Assistant Engineer **W. W. White**, U. S. Navy.

HONORARY MEMBERS.

(EX-OFFICIO.)

The Secretary of the Navy.

The Assistant Secretary of the Navy.

Chief of Bureau of Steam Engineering:
Geo. W. Melville, Engineer-in-Chief, U. S. N., Navy Department.

Ex-Chiefs of Bureau of Steam Engineering:
Chas. W. Copeland, Consulting Engineer, 24 Park Place, New York.
Chas. H. Haswell, Consulting Engineer, 42 Broadway, New York.
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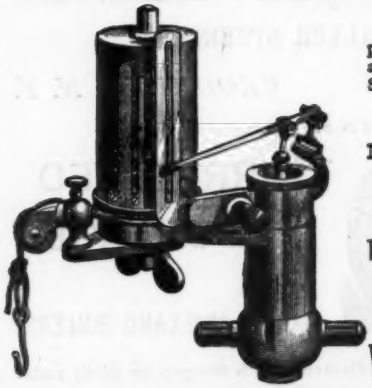

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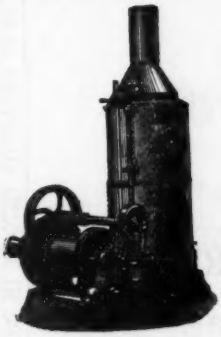

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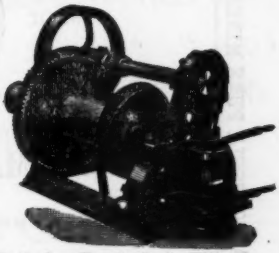

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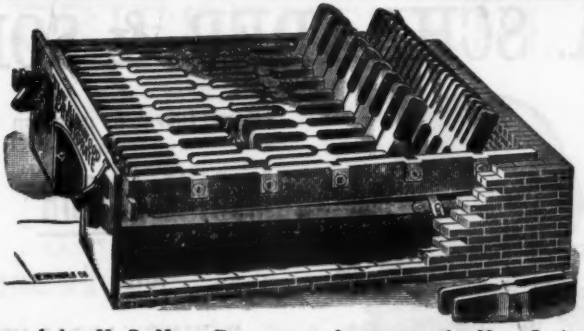

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